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on Fundamentals of Friction, Lubrication, Wear,
and their Control in Industry

WEAR

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ON FUNDAMENTALS OF FRICTION, LUBRICATION, WEAR,
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EDITORIAL

Volume Properties, Friction and Wear

The correlation between elastic and plastic properties of engineering materials and their friction and wear resistance has been elucidated further in recent work. The relationship between rolling force and material specific damping capacity has been studied by DRUTOWSKI[1]. Starting from TABOR's concepts of rolling friction[2], it is shown that energy losses of a metal ball rolling on a metal plate are essentially due to hysteresis. The origin of the latter can be traced to the elastic strain of ball and plate and also, under certain conditions of load, to plastic deformation of the ball. This study was initiated in order to evaluate materials with optimum properties for instrument ball bearings, where bearing torque is critical.

The friction trace shows, as usual, a very complicated pattern. Closer analysis reveals that the individual peaks depend not only on surface roughness but also on orientation of material in the hardened plate. This latter effect has also been demonstrated for rolling friction by HALAUNBRENNER[3], who used a different experimental approach, with a single crystal of NaCl as a base.

The list of factors influencing the fluctuations of a friction trace is a long one. It is therefore of practical significance to the engineer that *average* friction force in rolling is a simple function of a volume property, *i.e.* the specific damping capacity of solids.

Cool running tyres are the ideal of automotive engineers. Their demand is quite a headache to rubber compounders, who must compromise between an acceptable level of generated heat and "super abrasion" stress–strain properties of a reinforced rubber. That high elastic hysteresis contributes not only to riding comfort but also to wear resistance comes rather as a surprise. We learn this fact from a study on the wear of slipping wheels by SCHALLAMACH AND TURNER[4]. They show that sliding under conditions of small angles of slip is confined to the rear area of contact between wheel and road. Wear by slip *decreases* with increasing elastic hysteresis of the wheel. These experiments confirm, as was already known empirically, that wear testing in the laboratory cannot reproduce conditions of service unless the kinematics of a rolling wheel is taken into account. It now becomes clearer why a highly damping rubber such as butyl is an interesting raw material for the tyremaker, notwithstanding the fact that such tyres exhibit a comparatively high rolling resistance.

The intense, but still empirical search[5,6,7] for seal and bearing materials, resistant to extreme conditions of speed and temperature, stresses the need for more basic information on primary processes in sliding friction. An analysis of present knowledge on work-hardening of metals in the sliding surface has been given previously by RUBENSTEIN[8]; it is supplemented by a short note in this issue (p. 150). The possibilities of a *deductive* approach are probably exhausted with this lucid elaboration of a

general theory. Further critical experiments are now needed to determine the influence of surface geometry, localized stresses and boundary lubrication on friction force.

Flow stress and erosion resistance of ductile materials are correlated by a semi-quantitative theory developed by FINNIE (p. 87). In this analytical approach two apparently quite unrelated fields of engineering technology are interlinked: production engineering and wear prevention. The idealized picture of the erosion process is that of surface cutting by small particles, known to occur in milling and grinding operations. FINNIE points out that high strain rates, large total strains and small volumes of the affected area prevail in micromilling as well as in erosion. Flow stresses in surface layers are therefore assumed to be similar in both operations. Consideration of particle shape and of the kinematics of collisions leads to a reasonable prediction of erosion resistance of metals.

A different approach is used by FINNIE to analyse the erosion resistance of materials that lack plasticity—at high strain rates—and therefore fracture in brittle fashion. Rubbers and glasses come into this second category. Assuming only Hertzian stresses to occur, the theory roughly predicts conditions of primary crack formation. Erosion performance, however, still has to be established by trial and error. The same is true for the numerous, recent experiments in which particles, ranging from rain drops to metal spheres, are fired against a solid wall[9]. Additional information on the sudden decrease of damping capacity which *all* solids experience at low temperatures, would presumably allow also a prediction of brittle properties at very high strain rates and ambient temperatures. A general theory of impact comminution already exists[10]; it is, in fact, the counterpart of a future erosion theory of brittle solids.

Delft, January 1960 G. SALOMON

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[3] J. B. HALAUNBRENNER, *Wear, 2* (1958/59) 423.
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[5] F. M. SAUER, *OTS Report PB 131963*; for abstract see *Wear, 3* (1960) 76.
[6] L. B. SIBLEY *et al.*, *OTS Report PB 151483*; for abstract see *Wear, 3* (1960) 75.
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[8] C. RUBENSTEIN, *Wear, 2* (1958/59) 85.
[9] Reports from Royal Aircraft Establishment (Gt. Brit.); for abstracts see *Wear, 2* (1958/59) 498.
[10] H. RUMPF, *Chem.-Ing.-Tech., 31* (1959) 323; for abstract see *Wear, 3* (1960) 73.

EROSION OF SURFACES BY SOLID PARTICLES

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SUMMARY

The amount of surface material eroded by solid particles in a fluid stream depends on the conditions of fluid flow and on the mechanism of material removal.

The paper first discusses some aspects of the fluid flow conditions which may lead to erosion and then analyses the mechanism of material removal for ductile and brittle materials.

For ductile materials it is possible to predict the manner in which material removal varies with the direction and velocity of the eroding particles. The numerical magnitude of the erosion cannot be predicted with accuracy but does correlate with data from metal cutting tests.

For brittle materials the conditions leading to initial cracking are deduced and ways of predicting the material removal are discussed. It was not found possible to develop an analysis as detailed as that for ductile materials.

In addition, the influence of the properties of the abrasive particle on erosion is briefly considered.

ZUSAMMENFASSUNG

OBERFLÄCHENEROSION DURCH FESTE TEILCHEN

Die Erosion von Oberflächen, durch feste Teilchen in einer strömenden Flüssigkeit, hängt von den Bedingungen des Fliessens und von der Mechanik des Abtrennens des Materials ab.

Zunächst werden einige Gesichtspunkte der Fliessbedingungen, die zu Erosion führen können, besprochen, dann wird eine Analyse der Mechanik des Abtrennens für plastisches und sprödes Material gegeben.

Für plastisch deformierbares Material kann die Änderung des Trennprozesses als Funktion der Richtung und Geschwindigkeit erodierender Teilchen vorausgesagt werden. Der Zahlenwert der Erosion ist nicht genau vorauszuberechnen jedoch besteht eine Beziehung zu den Messergebnissen, die beim Abdrehen von Metallen erhalten wurden.

Bedingungen die zur anfänglichen Rissbildung von spröden Materialien führen werden abgeleitet und Wege, um die Abtrennung vorauszusagen, werden besprochen. Jedoch erwies sich eine ebenso eingehende Analyse wie für plastisch deformierbare Materialien nicht als möglich.

Schliesslich folgt eine kurze Betrachtung über den Einfluss der Teilcheneigenschaften auf den Erosionsvorgang.

INTRODUCTION

In many applications a surface is attacked by solid particles entrained in a fluid stream. This type of wear is generally described as erosion. Probably the most important erosion problems which occur in industry are those connected with the equipment used in the catalytic cracking of oil. However, erosion is also a continuing problem in such units as coal turbines, hydraulic turbines, and coal hydrogenation equipment. While usually considered undesirable, erosion has useful application in such processes as sand blasting, abrasive deburring and the erosive drilling of hard materials.

Many studies may be found in the literature of specific erosion problems and their practical solution[1]. However, the basic principles of this industrially important process have received little attention and are not generally appreciated or understood. There is, in addition, little complete experimental data in the literature, which further limits the possibility of predicting erosion under new and untried circumstances. In particular, the particle velocity, as distinct from the fluid velocity, does not seem to have been measured in erosion tests.

It seems clear that an understanding of erosion may be divided into two major parts. The first part involves a determination from the fluid flow conditions of the number, direction, and velocity of the particles striking the surface. With such information available, the second part of the problem is a calculation of the amount of surface material removed. The first part of the problem is, basically, one of fluid mechanics, and its detailed treatment lies outside the scope of this article. However, some aspects of the particle motion in the fluid will be mentioned. In particular, the problem of predicting particle velocities in erosion tests will be considered.

Turning back to the second part of the problem, that of predicting the amount of material removed, we can see that it is unlikely that one mechanism of material removal will apply to all materials. For the purpose of a first analysis, it seems convenient to limit the discussion to two main types of material behaviour, ductile and brittle. Ductile materials will undergo weight loss by a process of plastic deformation in which material is removed by the displacing or cutting action of the eroding particle. In a brittle material, on the other hand, material will be removed by the intersection of cracks which radiate out from the point of impact of the eroding particle. Many engineering materials such as annealed low-carbon steel, copper, aluminum, glass and ceramics fall rather obviously into one of these two categories. While it seems clear that many other materials will not be so easily categorized, at least this method forms a simple starting point for an erosion analysis. Later in the paper we will consider the mechanism of material removal for ductile and brittle materials in more detail and will also examine experimental data. These erosion test data were obtained with a specially constructed erosion tester in which the various variables such as particle velocity and direction relative to the surface may be measured and controlled. A description of the tester has been given by the writer[2].

LITERATURE SURVEY

A survey of the extensive literature on erosion, up to 1946, has been given by WAHL AND HARTSTEIN[1]. Although there is a wealth of practical information in the many references discussed in this survey, there is little direct information on the mechanism by which material is removed during erosion. Studies of erosion from a fundamental point of view appear to have originated in Germany. SIEBEL AND BROCKSTEDT[3] eroded plates with a stream of quartz sand directed perpendicular to the surface and found that the weight loss of hard or soft steels, as well as alloyed

or unalloyed steels, was very much the same. These results were not in agreement with the practical experience that pipes of hard steel showed considerably longer life than pipes of soft steel when transporting abrasive material. WELLINGER and co-workers[4-8] carried out erosion tests with different angles of impingement and cleared up the contradiction by showing that the relative merit of different materials, for erosion resistance, could change as the angle of impingement changed. These authors have chosen to express erosive wear as (mm^3 removed/cm^2 surface) and hence, before comparing their wear figures as a function of jet angle with those of authors who have tabulated total weight loss, we must allow for the variation of eroded area and divide by sin α, where α is the angle at which the jet is inclined to the surface (see Fig. 1). On this basis, they find for a steel St. 37 (Vicker's hardness 125 kg/mm^2) that the maximum weight loss occurs when the jet is at angles of about 30° to the surface, while for the harder steel St. 70 (Vicker's hardness 840 kg/mm^2) the maximum weight loss occurs at jet angles close to 90° (perpendicular impact). A number of investigators, prior to WELLINGER, had reported such measurements as the angle for maximum erosion under certain test conditions. However, the work of WELLINGER and his colleagues appears to mark the first attempt to gather test data on the process of erosion, under a wide variety of controlled conditions. Following HOLTEY[9], these authors attribute the weight loss in erosion to two processes which are called "rub" and "shock" wear. The wear curves are divided into these components, and it is shown that soft steels are superior under "shock" conditions while hard steels are best for "rubbing" wear. No further correlation of the data with the physical properties of the materials was attempted nor was an explanation given for the mechanism of material removal. As already mentioned, WELLINGER's experiments confirmed the practical experience that soft steel pipes wore more than hard ones in abrasive service. On the other hand, in tests where more nearly perpendicular erosion was involved. the softer steel showed to advantage and this, also, is confirmed by practice. Similar results can be found in the work of KASHCHEEV[10], who eroded a range of copper–aluminum alloys at different angles.

With this background of work in erosion, it appears that further progress in understanding and applying erosion data depends upon an improved fundamental understanding of the processes involved in material removal. At the same time, the effect on erosion of changes in the flow conditions should also be understood.

FLOW CONDITIONS

The erosion of a surface by abrasive particles in an inert fluid should depend on the number of particles striking the surface, their velocity, and their direction relative to the surface. These quantities are largely determined by the flow conditions and many practical examples may be found where a change in flow conditions has greatly increased or decreased erosion. In units where the flow direction changes fairly rapidly (turbine blading, valves, pipe bends, etc.) erosion is usually considerably more severe than in a straight run of piping. Local turbulence due to a roughened

surface or misaligned parts may greatly increase erosion. Probably for this reason, it is often found that erosion problems increase rapidly during the life of a unit working under erosive conditions. In pumps, and other fluid-handling units, localized erosion may often be attributed to secondary flows.

WAHL AND HARTSTEIN[1] describe a number of German patents on the protection of bends in pipes carrying abrasive materials. In one device air is blown in tangentially, along the outside of the bend. Tests apparently showed twelve times the life of a normal bend. It has been said that a logarithmic spiral ("highway curve") should be the ideal shape for a bend in a pipe carrying abrasive material. Although intuitively it would seem desirable, for uniform erosion, to have constant rate of change of curvature between two straight runs of pipe, this has not yet been confirmed by experiment. WAHL AND HARTSTEIN also discuss the importance of exact alignment of pipes carrying abrasive material. References are also given in this monograph to work on the erosion of different types of fans and nozzle blades.

The problem of particle impaction onto different shapes was apparently first studied in Germany in 1931 in connection with the collection of smoke and dust particles. More recently, the problem of particle impaction has also been studied in connection with the icing of aircraft, first by TAYLOR[11] in 1940, and later by many authors with respect to specific airfoils, inlet ducts, and other shapes. Often Stokes' Law has been used to determine particle resistance and occasionally the experimentally determined drag coefficients have been used. The only case for which a direct analytical solution appears possible is that of flow onto a normal wall, with Stokes' Law assumed for the particle drag, any other case requiring a stepwise solution. If the fluid velocities are known and the drag forces on the particles are calculated using experimental drag coefficients (for example, as given by LANGMUIR AND BLODGETT[12] for spheres), then the impaction of particles onto a given obstacle may be computed numerically.

SMITH[13] has studied particle impaction in connection with the fouling of turbine blades and this work is also of interest in studies of blade erosion by particles entrained in the gas stream. He considered impaction on the nose of the blade as well as the concave side of the trailing edge and gave approximate catch efficiencies as a function of the dimensionless impact number, this latter quantity basically being the ratio of the particle drag to its inertia. An indication can be obtained from this report of the size of particles which should be removed from a stream to avoid erosion of turbine blades. Factors which apparently have not yet been considered are the radial motion of the particles in the blading and the effect of multistages.

BERGERON[14] studied particle trajectories in centrifugal pump runners and developed similitude principles for predicting pump erosion from model tests. BERGERON's work refers to the erosion of the convex face of the runner, since the erosion of the leading edge by "impact" is considered to be negligible in comparison with the erosion by "sliding" on the blade surfaces.

Since the relative erosion resistance of different materials depends on the angle at which a surface is being attacked, it is of value to be able to assign an approximate

angle to the impaction occurring under different flow conditions. Some units such as partially open slide valves and sharp bends may be hit by particles at angles close to 90° (perpendicular to the surface) but in most cases of pipe flow the angle must be considerably smaller. WELLINGER AND UETZ[7] estimated this angle to be 5° for their particular test conditions of pipe flow by comparison with plate tests. We would thus expect soft carbon steels to show to advantage in the former type of unit and harder carbon and alloy steels to be superior for straight pipes. This is in good qualitative agreement with practice.

The velocity and direction of an abrasive particle in a fluid are not necessarily the same as those of the fluid. For this reason the influence of particle velocity and particle size on erosion is not at all clear. Experimenters have usually expressed weight loss as a function of fluid pressure or fluid velocity and the relationship of these quantities to particle velocity depends on the particle size and shape as well as the particular flow conditions of the experiment.

As a guide to the actual particle velocities in experiments for which fluid velocities have been reported, we will estimate the distance traveled by a spherical particle, introduced at rest into a gas stream, before it acquires a given percentage of the gas velocity. If take we as an example a particle of 0.01 in. diam. ($\approx$ 60 mesh) and an air stream of 500 ft./sec (152 m/sec) at 32°F (0°C) and atmospheric pressure, the initial Reynolds number for the particle is $R = 2920$. Tabulated values of the drag coefficient for spheres as a function of Reynolds number show that in this range the drag coefficient may be taken as independent of Reynolds number. As the particle velocity increases the effective Reynolds number decreases and the assumption of constant drag coefficient becomes poorer. However, down to about $R = 200$, the assumption of constant drag coefficient will be sufficiently accurate for our purposes.

The equation of particle motion is then

$$\frac{4}{3}\pi r^3 \varrho_p \frac{dV}{dt} = \frac{C_D}{2} \varrho_a \pi r^2 (U-V)^2$$

where r, V, and ϱ_p are the particle radius, velocity, and density, U and ϱ_a are the air velocity and density, and C_D is the drag coefficient. Putting

$$K = 3C_D \varrho_a / 8 r \varrho_p$$

and noting that $dV/dt = 0$ at $t = 0$, the equation is

$$-d(U - V)\, dt = K(U - V)^2$$

Integrating and putting $V = 0$ at $t = 0$

$$V/U = UKt/(1 + UKt)$$

or

$$1/(U - V) - 1/U = Kt$$

Integrating again leads to an expression for the distance (x) traveled by the particle in a given time

$$x = \frac{1}{K}[UKt - \ln(1 + UKt)]$$

For a particle 0.01 in. diam. of specific gravity 3.0 in air at 32°F and atmospheric pressure and $C_D \simeq 0.39$ (ref.[12]) we have $K = 0.152$ ft.$^{-1}$.

Taking a typical value for the air velocity in an erosion test as say 500 ft./sec (152 m/sec) we find that the particle must travel 3.8 ft. (1.6 m) before reaching 300 ft./sec (91 m/sec). In most erosion testers the particles travel a fairly short distance in the high-velocity air before reaching the specimen, and their velocity will thus be only a fraction of the air velocity. This uncertainty of particle velocities makes it impossible to correlate the results of different erosion experiments if only the air velocity is reported. The particle velocity depends on its size and for this reason the effect of particle size on erosion is uncertain.

DUCTILE MATERIALS

During erosion of a ductile material a large number of abrasive particles strike the surface. Some of these will land on flat faces and do no cutting, while others will cut into the surface and remove material in a manner somewhat similar to the tooth of a milling cutter or the grains on a grinding wheel. As a starting point, we will consider the motion of a single rigid abrasive particle which strikes a ductile material in such a way as to displace or cut away part of the surface. After the behavior of the single rigid particle has been explored, we will discuss the influence of multiple impacts and non-rigid abrasive grains.

The crater or scratch left by an impacting particle is the first clue to the manner in which erosion occurs. Normally, erosion produced by particles striking the surface at shallow angles (low values of α in Fig. 1) is the condition of greatest interest and for this case the craters have length/depth ratios of the order 10:1. This leads to the idealized picture of cutting, shown in Fig. 1, in which only the leading face of the particle contacts the surface. For particle impaction at higher values of the angle α the picture shown in Fig. 1 will not be correct since both faces of the particle will meet the surface.

Our goal is to solve the equation of motion of the particle in Fig. 1 and thus by knowing the path it follows through the surface to predict the material removal. A detailed derivation of the equations involved in this approach has been given by the writer[15]. In the present discussion the assumptions made in the previous analysis will be restated and the results quoted. The predictions will then be compared with experimental data.

Before the equations of motion of the particle in Fig. 1 can be solved, the forces acting on the particle must be known. In the absence of an exact solution it will be assumed that the ratio of the vertical force component on the particle face to the horizontal force component has a constant value K. The assumption should be sufficiently accurate if a geometrically similar configuration is maintained throughout the period of cutting. This will be true only if the particle rotation during cutting is fairly small and such can be shown to be the case. It is then reasonable to assume that the ratio of the depth of contact l in Fig. 1 to y_t the depth of cut has a constant

value ψ. Further assumptions required for a tractable analysis are that the particle cutting face is of uniform width, which is large compared to the depth of cut (plane strain), and that a constant plastic flow stress p is reached immediately upon impact. Having made these assumptions, it is possible to write down and solve the equations of motion of the particle in the x, y and Φ directions and determine the coordinates of the particle tip x_t, y_t as a function of time. The volume of surface material removed by the particle is then taken as the product of the area swept out by the particle tip and the width of the cutting face.

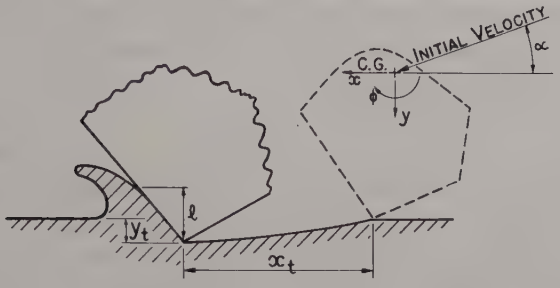

Fig. 1. Idealized picture of abrasive grain striking a surface and removing material. Initially the velocity vector of the particle's center of gravity makes an angle α with the surface.

The only problem now is that of defining the time at which cutting ceases. Obviously, at low values of the angle α in Fig. 1, the particle comes down, cuts out part of the surface and then leaves again. Cutting stops when the particle tip leaves the surface (*i.e.*, $y_t = 0$). At higher angles the horizontal motion of the particle tip may cease before it leaves the surface. Cutting then stops when $dx_t/dt = 0$.

Integrating over the cutting period leads to the following simple expressions for the volume of material, Q, removed by a single abrasive grain of mass, m, and velocity, V.

$$Q = \frac{mV^2}{p\psi K}\left(\sin 2\alpha - \frac{6}{K}\sin^2\alpha\right) \quad \text{if } \tan\alpha \leqslant \frac{K}{6}$$
$$Q = \frac{mV^2}{p\psi K}\left(\frac{K\cos^2\alpha}{6}\right) \quad \text{if } \tan\alpha \geqslant \frac{K}{6} \qquad (1)$$

The two expressions predict the same weight loss when $\tan 2\alpha = K/6$ and the maximum erosion occurs at the slightly lower angle given by $\tan 2\alpha = K/3$. The first equation, which applies for lower angles, corresponds to the case in which the particle leaves the surface while still cutting. The second equation, which applies to higher angles, corresponds to the case in which horizontal motion of the particle tip ceases while cutting.

The manner in which the predicted volume removal of eqn. (1) varies with angle α may be seen in Fig. 2 for the case of $K = 2$. The value of K, the ratio of vertical to horizontal force components on the particle is not amenable to direct measurement.

However, force measurements during two rather similar processes, scratch hardness testing[15] and grinding[16], show an overall value of K very close to 2 for angular abrasive grains. It seems reasonable to use this value of K in the erosion analysis when considering erosion by angular grains.

Before going any further it is of interest to compare the predicted volume removal–angle relation for a single grain with data obtained from tests in which very many grains (10^5–10^6 per cm^2) strike a surface. Experimental data for the erosion of aluminum, copper, and steel by angular silicon carbide grains are also shown in Fig. 2. The scales of the data points for the three metals are adjusted so that the maximum erosion, in each case, is the same as the predicted value. Very similar curves are obtained for erosion by "block-like" aluminum oxide particles.

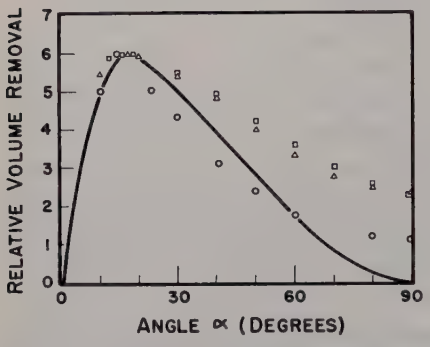

Fig. 2. Predicted variation of volume removal with angle for a single abrasive grain. Experimental points for erosion by many grains (△ copper, □ SAE 1020 steel, o aluminum) are plotted so that the maximum erosion is the same in all cases.

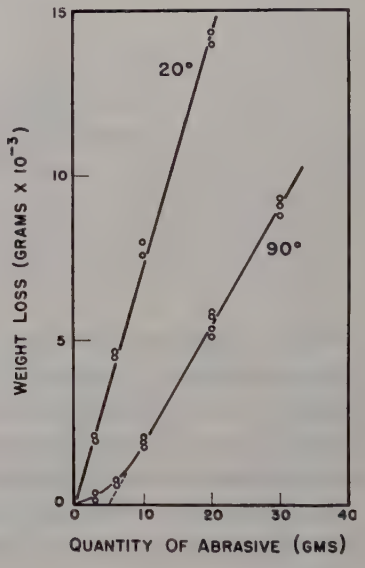

Fig. 3. Variation of weight loss with quantity of abrasive. Material, annealed SAE 1020 steel; abrasive, 60 mesh silicon carbide; Velocity, 250 ft./sec (76 m/sec)

Fig. 2 shows that the predicted variation of material removal with angle for a single grain has the same general form as experimental data for erosion by many grains. In particular, the predicted angle for maximum erosion agrees closely with experiment. Such agreement indicates that the single grain analysis may be applied to predict erosion by a multitude of grains, at least for low-angle erosion. The most unsatisfactory part of the analysis is that it underestimates erosion at high angles and in particular predicts no erosion at 90°. The disagreement with experiment is even more severe if we recall that Fig. 1 shows contact on only one face of the particle, while at high angles both faces may strike the surface. This will result in even lower predicted erosion at high angles.

The principal reason for the variation between prediction and experiment in Fig. 2 appears to be the fact that the analysis assumes that the particle strikes a smooth surface while actually the particle sees a surface which has been roughened by previous particles. At low impact angles (say $\alpha \leqslant 30°$) the "slice" taken by a

particle out of a smooth surface is relatively shallow and the surface shape is not greatly disturbed. However, at larger angles the depth/length ratio of the impression left by a particle on a smooth surface is relatively much greater. Subsequent particles then strike the eroded surface at a wide variety of "true" angles. This explanation is confirmed by direct observation of the surface and by tests, such as those shown in Fig. 3, in which increasing amounts of abrasive are used to erode the same area. At the low impact angle (20°) the weight loss produced by a given quantity of abrasive is seen to be independent of the previous erosion. At the high angle (90°) the material removal due to a few grams of abrasive is at first very low and then increases with prior erosion. By visual inspection, the surface roughness is seen also to increase during this period.

Empirical corrections, based on the type of experiment shown in Fig. 3, may be used to correct the predicted curve of Fig. 2 for erosion at angles near 90°. For erosion at low angles, which is very often the case of interest in practice, eqn. (1) will be used without correction. To extend the analysis for the single grain to multiple particle impact the mass of a single abrasive grain in eqn. (1) will be replaced by M, the mass of many grains. At the same time it must be recognized that many particles will not cut in the idealized manner but rather will strike on flat faces and do no cutting. Guided by inspection of the erosion craters due to a known number of abrasive grains we rather arbitrarily take 50% off the predicted erosion to allow for the fact that many particles will not be as effective as the ideal particle*.

The only remaining unknown is the ratio ψ (l/y_t in Fig. 1) and by analogy to metal-cutting experiments[17] a value of $\psi = 2$ is selected. Finally, free from further assumptions, we may write the volume removal due to a mass M of angular abrasive grains as

$$\begin{aligned} Q &\approx \frac{MV^2}{8p} [\sin 2\alpha - 3 \sin^2 \alpha] \quad && \alpha \leqslant 18.5° \\ Q &\approx \frac{MV^2}{24p} \cos^2 \alpha && \alpha \geqslant 18.5° \end{aligned} \tag{2}$$

with the limitation that for angles above say 45° eqn. (2) greatly underestimates the weight loss.

In particular, for angles close to that for maximum erosion (15–20°) the volume removal is given by

$$Q \approx 0.075 \left(\frac{MV^2}{2}\right) \frac{1}{p} \tag{3}$$

That is, 7.5% of the particle kinetic energy divided by the flow pressure gives an estimate of the volume removal.

Velocity dependence of weight loss

Turning first to the variation of weight loss with velocity predicted by eqn. 2, we find that this is in good agreement with experiments on several metals. Fig. 4 shows experimental data for the weight loss of an annealed low-carbon steel eroded

* We will discuss later the effect of varying the type of abrasive particle.

by silicon carbide grains. The line $W \sim V^2$ is seen to give a good fit to the data. For these experiments the particle velocities were measured from photographs taken with a high-speed, double-flash, high-intensity light source. Particle velocities have apparently not previously been measured in erosion tests.

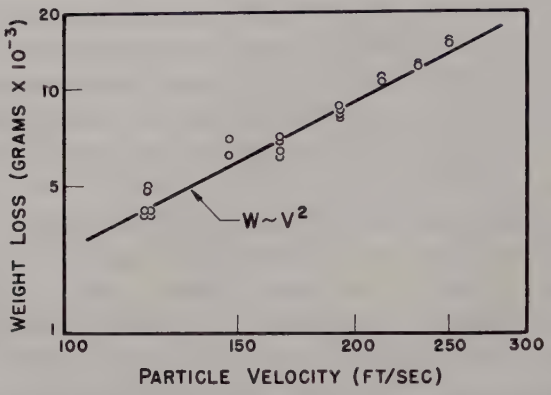

Fig. 4. Weight loss of annealed SAE 1020 steel as a function of velocity for erosion at $\alpha = 20°$ by 20 g of 46 mesh silicon carbide grains.

At sufficiently low velocities we would expect that progressively more particles would produce only elastic stresses in the surface and hence the weight loss at low velocities would decrease more rapidly than predicted by the expression $W \sim V^2$. As might be expected, this departure from the predicted relation was found to occur at higher velocities for sphere-like particles than for angular particles. At a particle velocity of 120 ft./sec (36 m/sec) it was found that hardened spherical steel shot eroding an annealed steel produced much less erosion than would be predicted from measurements at higher velocities. With angular silicon carbide particles as the abrasive no such effect was detected at this velocity.

In another set of tests it was found, for both spherical and angular abrasives, that the angle for maximum erosion was not influenced by particle velocity.

Numerical prediction of weight loss

In attempting to apply eqns.(2) or (3) to predict material removal there is a problem in choosing the appropriate value of the flow stress p. If it is to be obtained from another type of test, then the test conditions must be comparable to those occurring during erosion. Erosion involves very high strain rates, large total strains and very small test specimens (*i.e.*, the depth of cut). Since these three factors all tend to increase the flow stress it seems likely that erosion will be characterized by considerably greater flow stresses than are obtained in say tension or hardness tests.

From test data for an annealed low-carbon steel (SAE 1020) eroded by silicon carbide grains at 20° we find, from eqn. (3), values of p between 350,000 and 400,000 p.s.i. (24,000 to 28,000 kg/cm²). Although these stresses are about three times the true stress

at fracture in a tension test on the same material, they may not be unreasonable when the factors which may influence the flow stress are considered. Further justification for such high predicted flow pressures may be found in the literature. Micromilling[18] is a process in which a thin layer of known shape is removed from a metal surface. A flow stress may be calculated from the measured forces and the known geometry of the cut. In one set of tests, reported by BACKER, MARSHALL AND SHAW[18] on a SAE 1112 steel the depths of cut and cutting velocities were of the same order as those in the erosion tests just cited. Analysis of the micromilling data gave flow stresses between 300,000 and 400,000 p.s.i. The agreement between the micromilling and erosion test data is quite encouraging and indicates that data from other tests could be used to estimate material removal in erosion of ductile metals.

However, in view of the present uncertain nature of the flow stress in erosion, it may be preferable to establish this quantity by an erosion experiment, with the abrasive and surface material of interest before using eqn. (2) to predict the influence of angle and velocity. This procedure has been followed, for example, in predicting turbine blade erosion.

Influence of the abrasive

In the preceding discussion very little attention has been paid to the abrasive particle beyond specifying that it was rigid and angular. We will now consider some of the ways in which three properties of the particle, shape, hardness and strength, may influence the erosion of ductile metals.

It is clear that sharp angular particles will produce more erosion than sphere-like particles if all other properties are the same. This is confirmed by one of the earliest tests in the erosion literature[19] in which it was noted that for sandblasting at 45° to the surface "sharp" sand gave four times the wear of "round" Ottawa sand. The parameter in the analysis which may change with changing particle shape is K (the ratio of vertical to horizontal force components on the particle). It seems reasonable that K will increase as the particles become less angular and more nearly spherical. An increase in K increases the angle for maximum erosion predicted by eqn. (1) and decreases the surface weight loss at this angle. The predicted angle for maximum erosion is not too sensitive to K, and hence presumably not too sensitive to particle shape. (For $K = 1.5$, $\alpha_{max} = 13°15'$; $K = 2.0$, $\alpha_{max} = 16°51'$, $K = 2.5$, $\alpha_{max} = 19°44'$; and for an extreme case $K = 5.0$, $\alpha_{max} = 29°31'$.) This is confirmed by tests with angular silicon carbide particles and "blocky" aluminum oxide particles which showed essentially the same angle for maximum erosion ($\alpha_{max} \approx 16°$). On the other hand, in the extreme case of erosion by hardened spherical steel shot the maximum erosion was found at about 28°. Although the analysis, based on Fig. 1, is not strictly applicable to erosion by spherical particles the increase in the angle for maximum erosion confirms the trend that would be expected. The velocity dependence of the weight loss, as we have already discussed, is not influenced by shape except insofar as the spherical particles produce relatively less volume removal at lower velocities.

When the hardness of the abrasive is greater than that of the surface it seems unlike-

ly that variation in the abrasive hardness will be important. By the same argument the relative wear resistance of different materials should show less variation as the abrasive hardness increases. This is confirmed by the tests of WELLINGER[6] and RIDGWAY *et al.*[20].The analysis which has been presented is only strictly applicable when the abrasive is harder than the surface. When the abrasive is softer than the metal the wear would be expected to decrease greatly. In the extreme case, this behavior is noticed when soft organic materials such as walnut shells and rice husks are used to remove scale, rust, or other deposits while leaving the underlying metal virtually untouched.

The strength of the abrasive will determine the extent to which grains cut as a single unit (as in Fig. 1) or fracture internally, while cutting, and hence reduce the predicted material removal. A certain amount of abrasive breakage occurs with the grains used in the present tests. In an earlier section the predicted weight loss was divided by a factor to account for grains which do not cut in the idealized manner. This may be considered to allow for grains which fracture internally as well as those which strike on flat faces.

A final variable which has not yet been mentioned is the size of the abrasive particle. In several tests in the literature the particle size has been varied but the uncertainty in particle velocities has prevented quantitative conclusions about the effect of different sized particles. In the present tests the range of particle sizes was rather small and no effect of size was noticed. It may be that with smaller particles the increased ratio of surface to mass will alter the cutting conditions and lead to an increasing volume removal from unit mass of abrasive.

BRITTLE MATERIALS

We studied the erosion behavior of ductile materials by solving the equations of motion for a rigid abrasive particle interacting with the surface. The volume of material removed was then estimated from the particle trajectory. For brittle materials this simple approach is no longer possible. Writing the equations of motion will lead to expressions for the stresses between particle and surface. These may then be applied to predict the initial fracture in the material. However, once fracture occurs the material removal depends on the propagation of the fracture surface into the material and its interaction with other fracture surfaces. About all that appears possible, at present, in the way of understanding the erosion behavior of brittle materials is to examine the conditions leading to initial cracking and to examine the influence of particle velocity and direction on volume removal by experiment.

The simplest case we can consider is an elastic sphere striking perpendicularly ($\alpha = 90°$) onto a brittle material which remains elastic until fracture occurs. The equations for this problem are well known[21] and show that the maximum tensile stress in the material occurs at the surface in a radial direction around the periphery of the contact area. The magnitude of this maximum radial stress can be shown to be

$$\sigma_r = 0.187\,(1-2\mu_2)\,\varrho^{1/5}\,V^{1/5}\left(\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right)^{-4/5} \qquad (4)$$

where ϱ, V, μ_1, and E_1, are the density, velocity, Poisson's ratio and modulus of elasticity of the impacting sphere and μ_2, E_2 are corresponding quantities for the surface. One would expect the initial crack to form at right angles to the maximum tensile stress (*i.e.* around the circumference of the contact area). In practice, such a ring crack is observed in glass although it lies slightly outside the contact area[22]. The ring crack flares out below the surface to form the mantle of a cone. Fig. 5 illustrates the crack shape schematically.

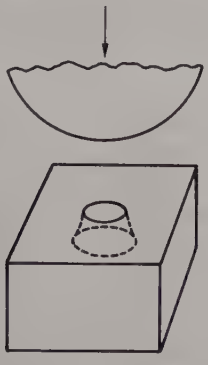

Fig. 5. Schematic drawing of the ring crack produced when a steel sphere strikes a section of a relatively large glass surface.

Equation (4) shows the influence of the elastic modulus and Poisson's ratio of the surface on the initial maximum stress. Although this has no direct bearing on the volume removal it does indicate that the erosion resistance of brittle materials will tend to increase with decreasing elastic modulus and increasing Poisson's ratio*.

To study the initial cracking of a brittle material, erosion tests were carried out in which increasing amounts of spherical steel shot were impacted against glass. In the very early stages of these tests each sphere produces a ring crack but no material is removed. Eventually, with an increasing number of impacts, the cone-shaped fracture surfaces, associated with the ring cracks, begin to intersect and material is removed. Once the original surface of the glass has been roughened by material removal it is no longer possible to observe the manner in which individual fracture surfaces are formed. Presumably the process of material removal continues to be basically the same as that observed for a smooth surface.

Since the ring crack plays a prominent role in, at least, the initial stages of material removal when glass is struck by spherical particles, it may be of interest to examine the influence of particle direction and velocity on ring cracking. The classical "Hertzian" equations[21] apply to perpendicular impact only ($\alpha = 90°$) and predict that the

* Although we have associated eqn. (4) with brittle materials, since they are usually idealized as behaving elastically up to the point of fracture, it applies more generally to any elastic material, for example rubber. The remarkable erosion resistance of rubber may be explained by its low modulus of elasticity and hence, from eqn. (4), the low tensile stresses which it experiences. In addition the large Poisson's ratio (nearly 0.5) of rubber also reduces the maximum tensile stress predicted by eqn. (4).

diameter of the ring crack* d is related to the particle velocity by

$$d = (\text{constant})\ V^{0.4}$$

If it is assumed that surface shear stresses due to oblique impact may be neglected, and only the vertical component of velocity is considered, then the preceding relation may be written as

$$d = (\text{constant})\ (V \sin \alpha)^{0.4} \qquad (5)$$

To study this relation, impact tests were carried out in the erosion tester at a variety of angles and velocities. Fig. 6 shows the variation in crack diameter when only the

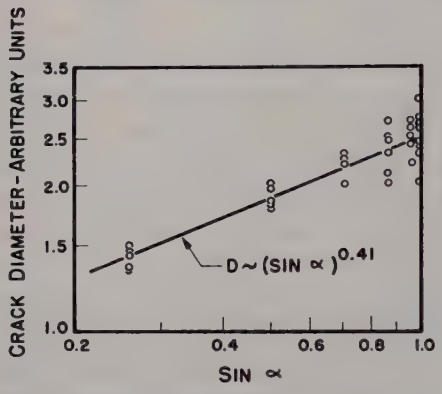

Fig. 6. Relation between diameter of cracking and angle of incidence for impact of hardened steel spheres, 0.023 in. diam., on an annealed glass.

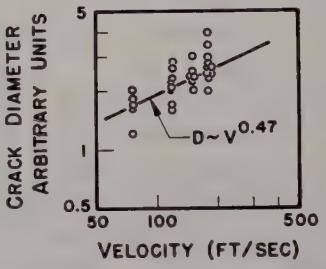

Fig. 7. Relation between diameter of cracking and impact velocity for an annealed glass struck by hardened steel spheres, 0.023 in. diam., at 90°.

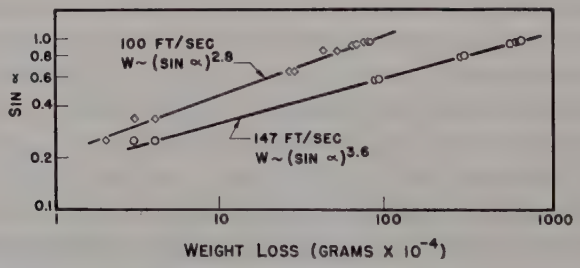

Fig. 8. Weight loss of an annealed glass as a function of angle when eroded by 53 g of 0.023 in. diam. hardened steel shot.

angle was changed. There is a lot of scatter, particularly at $\alpha = 90°$, but the general behavior is in accordance with eqn. (5). In many cases at lower angles the crack did not form a complete ring. The variation in crack diameter with particle velocity, for impact at $\alpha = 90°$, is shown in Fig. 7. There is again a considerable amount of scatter in the data but the over-all result does not disagree too badly with eqn. (5).

* Strictly speaking, it is the diameter of the contact area and not the crack which is related to the velocity. The difference in these diameters is small and in any event they are probably proportional to one another.

The two preceding test results, although limited in extent, indicate that velocity and angle enter together as $V \sin \alpha$ in determining the diameter of ring cracking. However, when the material removed by spherical particles is studied an entirely different situation is found. Fig. 8 shows the variation of surface weight loss as a function of Sin α for two velocities; and Fig. 9 shows the variation of surface weight

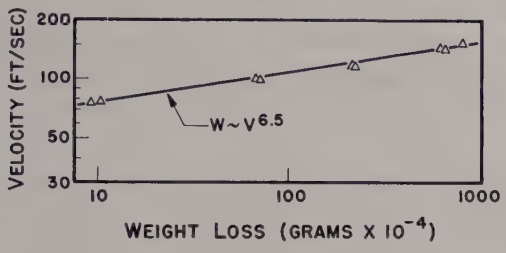

Fig. 9. Weight loss of an annealed glass as a function of impact velocity when eroded by 53 g of 0.023 in. diam. hardened steel shot.

loss as a function of velocity for $\alpha = 90°$. The empirical laws fitted to the data were $W \sim (\text{Sin } \alpha)^{2.8}$ for $V = 100$ ft./sec (30 m/sec), $W \sim (\text{Sin } \alpha)^{3.6}$ for $V = 147$ ft./sec (45 m/sec), and $W \sim V^{6.5}$ for $\alpha = 90°$. Tests on a different batch of glass and a porcelain at $\alpha = 90°$ gave the relation $W \sim V^{5.5}$ for the glass and $W \sim V^{3.3}$ for the porcelain. The principal conclusion to be drawn from such test data is that there is no very simple parameter which combines the effects of velocity and angle in producing material removal. From a practical point of view this is unfortunate since it greatly complicates a comparison of the erosion resistance of different brittle materials.

At present it does not seem possible to explain the preceding empirical relations for the weight loss. The origin and growth of fracture in brittle materials are not yet well understood and it appears likely that an ability to predict erosion in brittle materials will have to await progress in other fields. If a prediction of erosion is required there appears to be no satisfactory approach except that of testing under the specific conditions of interest.

Influence of the abrasive

The abrasive properties of importance in determining the erosion of ductile materials have already been discussed. It appears that these properties, size, shape, hardness and strength, will also be important, although probably to a different degree, in the erosion of brittle materials.

For perpendicular impact by a sphere the Hertz equations show that the maximum stress is independent of the particle size. However, TILLET[22] has shown that there is a definite "size effect" in that increasingly higher velocities are required to produce initial cracking with smaller particles. Hence for erosion by spherical particles one would expect the volume removal to decrease with decreasing particle size.

Tests on glass with angular silicon carbide grains and with spherical steel shot showed that, for the same mass and velocity, the carbide grains had less effect on

the erosion at 90° than on the weight loss at lower angles. It seems reasonable that the component of velocity tangential to the surface would have more effect with angular grains than with spherical ones.

CONCLUSIONS

The wear of a surface due to solid particle erosion depends on the motion of the particles in the fluid as well as the behavior of the surface when struck by the particles. These two parts of the problem are related in that a surface, roughened by erosion, may increase the fluid turbulence and hence accelerate the rate of material removal.

We have discussed the influence of the fluid motion on erosion rather briefly and have pointed out the difficulty in determining the conditions under which a particle strikes a surface.

For materials which may be characterized as ductile, it is possible to predict the angle of impact which leads to maximum erosion and the velocity dependence of erosion. The material properties which may be deduced from erosion experiments are shown to be in agreement with those obtained in another type of test (micromilling). Under a given set of erosion conditions the nature of the abrasive may strongly influence the material removal.

The influence of particle velocity and angle in the erosion of a brittle material is seen to be quite different from that of a ductile material. No analysis for brittle materials comparable to that obtained for ductile materials appears possible at the present time.

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TRANSIENCE OF THE STATE OF WEAR BY REPEATED RUBBING

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SUMMARY

In repeated frictional movement, wear rate varies with the rubbing distance, the rate being generally high at the beginning of the movement, falling with the rubbing distance, and finally reaching a constant value. The transition of these stages of wear is discussed. A special wear-testing machine, with which tests can be made on wear by virgin friction and by repeated rubbing, is used for both steel and copper against steel in dry and lubricated states. The effect on wear of surface roughness, of worn particles adhering to the surface, and of work-hardening of the surface is examined by means of various devices. It is concluded that the transition of the state of wear is mainly due to the gradual accumulation of worn particles on the rubbing surface but not to the formation of any particular surface layer or to variation of surface roughness.

INTRODUCTION

It is well known that the amount of wear caused by repeated rubbing of two solid surfaces does not increase in proportion to the rubbing distance. Although there are differences in degree of wear depending on the combination of materials, contact pressure, rubbing velocity and surface conditions, the wear usually increases very much at the beginning of the movement; then the wear rate gradually falls, eventually becoming constant. With a particular combination of material or under certain rubbing conditions, however, the wear rate is known to increase with the rubbing distance till it becomes constant. To distinguish the wear in varying stages of transition, that at the beginning is called the initial wear and that eventually reached the constant wear. When the wear rate in a part of a machine becomes constant, the part is said to be "run-in".

About the transient state of wear, there are a number of reports; according to one, the rubbing surface is gradually work-hardened by repeated frictional movement, thus diminishing the wear rate and improving the mechanical properties of the surface[1]. The accepted explanation of "running-in" of machine parts is that the gradual smoothening of surface roughness increases the contact area, which in turn reduces the actual contact pressure, and the wear rate is brought down in consequence. In this explanation, however, there still are many points to be accounted for. With some combinations of material or under some rubbing conditions, the constant wear is often observed to be very slight, sometimes even 1/100–1/10,000 of the initial wear, which fact cannot be explained by mere mechanical transformation such as work-hardening of surface. S. KOSAKA reported that, soon after the start of frictional movement, the

surface is smoothened to the constant state but the wear rate continues to vary and becomes constant long after the surface is smoothened[2]. Therefore the effect of roughness is considered not essential. HIRST AND LANCASTER in their recent study suggested that the transitional state of wear is associated with the formation of a surface film during sliding[3].

The purpose of the present work is to examine separately the formation of a particular layer by friction, the variation of surface roughness and the effect of adherence of worn particles upon the wear rate variation, and to study the fundamentals of the mechanism of transitional phenomena of wear.

FRICTION, VIRGINAL AND IN REPEATED RUBBING

Fig. 1 shows schematically the principal part of the apparatus used for the test and Fig. 2 the general view of the apparatus. A thin-walled pipe is turned on a turning

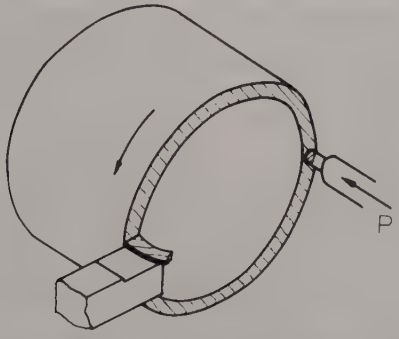

Fig. 1. The principle of the wear test in virginal rubbing.

Fig. 2. General view of the apparatus.

lathe, and its end face is machined by a cutting tool with feed. A small rod-shaped test piece is pressed against the end face opposite to the cutting tool and worn particles

adhering to the end face are scraped off by the machining after a half revolution. Thus the test piece is always rubbed by the virgin surface, the roughness of which does not vary so long as the cutting edge is not worn, which would affect the condition for examining the virgin friction. The friction in repeated rubbing is realized by simply removing the cutting tool and stopping the feed. If the material and dimensions of the test piece, contact pressure, rubbing velocity, etc., are the same in the above two settings, the wear by virgin friction can be compared with that by friction in repeated rubbing under gradual accumulation of worn particles and varying surface roughness.

For virgin friction tests, the end face of a pipe of annealed carbon steel (SAE1040) of 75 mm diameter and 5 mm wall thickness was machined with a cutting tool made of high speed tool steel with 20° side-rake angle and a small rod-shaped test piece of 4 mm diameter was pressed against the machined surface. Copper, carbon steel (SAE1040) 18-8 stainless steel, aluminum and 60/40 brass were used as the materials for the test piece. For repeated rubbing tests, the starting surface roughness was equal to that used for virgin friction tests with about 10 μ at the maximum for the pipe surface and about 1 μ for the test piece. The amount of wear on the test piece was measured using a chemical balance, taking due care not to disturb the worn particles left on the surfaces at the interruption of the test.

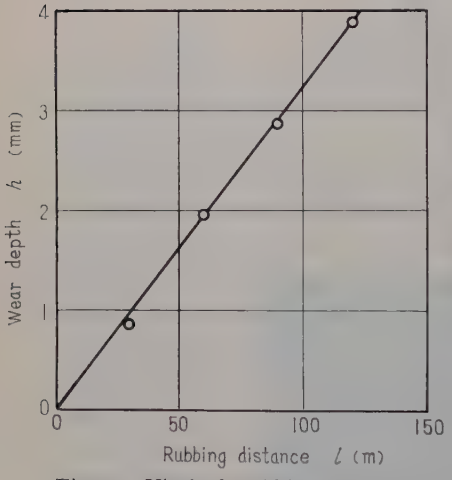

Fig. 3. Virginal rubbing test with copper. Large test piece: carbon steel SAE1040, Small test piece: copper. Contact pressure: 32 kg/cm². Rubbing velocity: 7.5 cm/sec (dry).

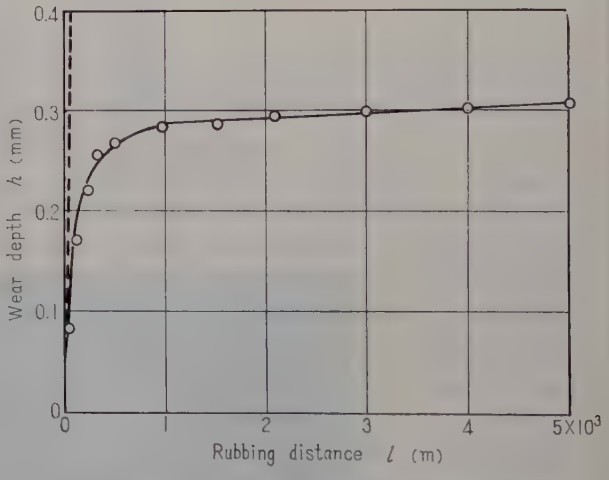

Fig. 4. Wear of copper in repeated rubbing. Same conditions as those in Fig. 3 except for the method of rubbing.

Figs. 3 and 4 show the results of tests on wear by virgin friction and by friction in repeated rubbing, respectively, with a test piece of copper rubbed against carbon steel under a constant pressure of 35 kg/cm² and rubbing velocity of 7.5 cm/sec in the dry state. From these figures we learn that the mode of wearing is fundamentally different in the two cases. Wear by virgin friction is extremely severe compared with

that by friction in repeated rubbing, the amount of wear increasing in proportion to the rubbing distance and the wearing continuing steadily. In repeated rubbing, on the other hand, the wear rate is initially high but falls rapidly; in the test it became almost constant after about 1000 m of rubbing. As the test piece is rubbed continuously, it is reasonable to conceive that some layer is being formed by friction on the surface, even in the virgin friction test. Results of the test, however, reveal that the mode of wearing remains unchanged with time, which suggests that such a surface layer, even if it happened to be formed, would have no effect upon the transient state of wear from initial to constant. The transience is observed only in repeated rubbing in which worn particles are gradually accumulating and its surface roughness is constantly changing; this proves that either the adherence of worn particles or the variation in surface roughness plays an important part in the transition process.

A test similar to the above was carried out with a pair of specimens of the same carbon steel. The result is given in Fig. 5, in which AB shows the variation of the

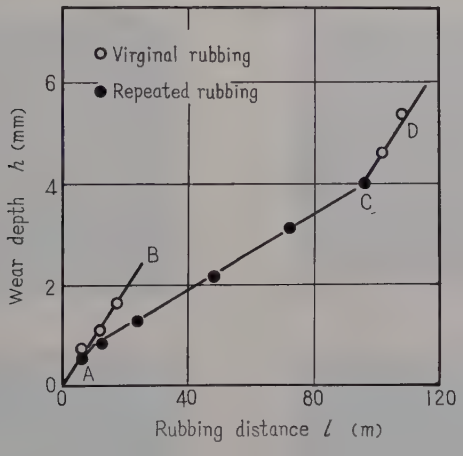

Fig. 5. Comparison of variations in the amount of wear by virginal and repeated rubbing. Carbon steel SAE1040. Contact pressure: 52 kg/cm². Rubbing velocity: 2.2 cm/sec (dry).

amount of wear in a virgin friction test and AC that in a repeated rubbing test. In the former, the wear is severe, increasing in proportion to the rubbing distance; in the latter, the increase in the amount of wear is gradually reduced and after a while the wear becomes constant, the tendencies parallel with those shown in Fig. 3 and Fig. 4. The test was continued from the point C in Fig. 5 ,with the cutting tool reset to freshen the surface for rubbing, reproducing the state of virgin friction. It is considered that by the time the process of continued rubbing reaches the point C on the figure the test piece will have a well-developed work-hardened layer on the surface. The result of this test shown by the line CD discloses not only the characteristics inherent to virgin friction but that the rate of wear is equal to that in the early test shown by the line AB, which fact confirms beyond doubt that the surface layer, if any, had no decisive relation to the wear process.

Tests were repeated with 18-8 stainless steel, aluminium and brass as materials for the test piece. Even though the hardness, tensile strength, yielding stress and work-hardening process were varied within a wide range, the results showed for all materials a tendency in wear process similar to that seen in Fig. 5.

VARIATION OF RUBBING SURFACE ROUGHNESS

The effect of surface roughness upon the transient state of wear was next studied.

With a copper test piece rubbing against a carbon steel pipe, the roughness of contact surfaces was measured by a surface roughness tester at intervals during the run of a repeated rubbing test. The roughness values thus measured on the two surfaces in the test shown in Fig. 4 are plotted against the rubbing distance in Fig. 6. From these two figures we learn that as the rubbing is repeated the wear rate on the test piece surface is reduced despite the fact that the roughness on the two surfaces is increasing, which is contradictory to commonly accepted conceptions. This means that, as KOSAKA pointed out, we shall have to consider the variation of the surface roughness as having no decisive relation to the transition of the state of wear from initial to constant.

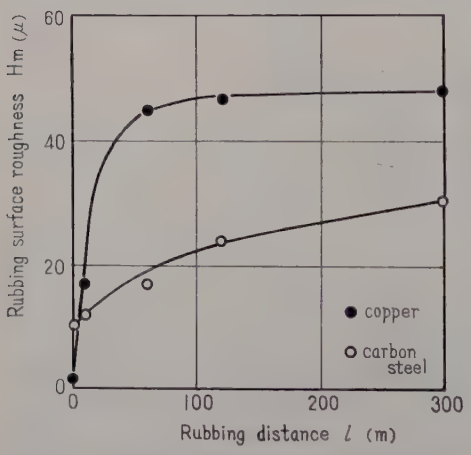

Fig. 6. Variations of roughness in repeated rubbing. Same conditions as those in Fig. 4.

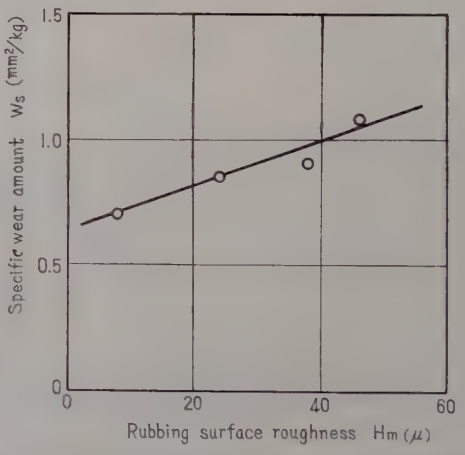

Fig. 7. Effect of rubbing surface roughness in virginal rubbing. Same conditions as those in Fig. 3.

Study was continued to ascertain further the relation of surface roughness to wear. To exclude the effect of the surface layer formed and of the accumulation of worn particles on the surface, the case of wear by virgin friction was taken up, and cutting tools of various side-rake angles were used to produce surfaces varying in roughness. With a copper test piece pressed against a surface of carbon steel pipe prepared in this way, the specific wear, *i.e.* the amount of wear per unit load per unit rubbing distance, in relation to the roughness is shown in Fig. 7, which is in accordance with the accepted conception that the rougher the surface, the more the wear. Accordingly, in the

case of repeated rubbing, if the surface roughness is being reduced with the increase of rubbing distance, this would hasten the falling-off of the wear rate. Nevertheless, the cause of the transition of wear to its constant state would seem to lie in the adherence of worn particles to the surface.

THE EFFECT OF ADHERED WORN PARTICLES

Fig. 8 shows the magnified photographs of the end face of a steel pipe repeatedly rubbed with a copper test piece. Worn particles of copper are seen accumulating in an ever-darkening discoloration on the surface as the rubbing distance increases. At 500 m of rubbing when the wear became almost constant, the layer of adhered worn particles that covered the entire surface was plainly visible. Such a layer of particles intervening between the rubbing and the rubbed surface would cause the wear rate to fall and by the time the accumulation reached its final constant state, the wear would also become constant.

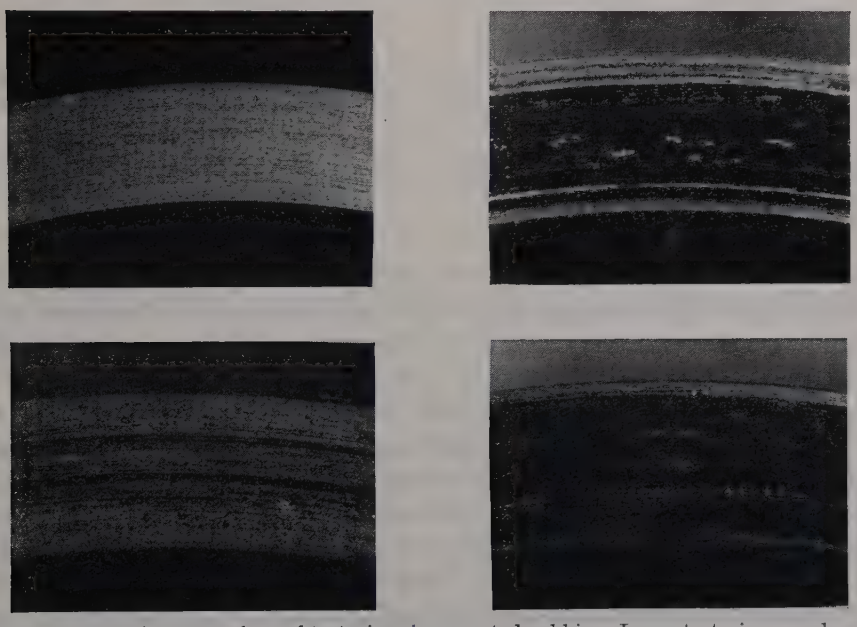

Fig. 8. Variations in the surface of test piece in repeated rubbing. Large test piece: carbon steel SAE1040. Small test piece: copper. Same conditions as those in Fig. 4.

In one test with repeated rubbing, the copper test piece was renewed, leaving undisturbed the worn copper particles adhering to the pipe during a 360 m run with the old test piece, thus starting the new test piece afresh on the old track to see how the layer of worn particles already formed on the rubbed surface would act on a fresh test piece, whether the wear process would progress exactly as that on the old one would have progressed if the test piece had not been replaced. The result is shown in Fig. 9

in which AB is by the old test piece, AC is by the new one. The wear on the new is far less than that on the old, and does not show its starting characteristics; AC can be correctly joined up as the extension BD to AB as though the new test piece had been in use from the very start.

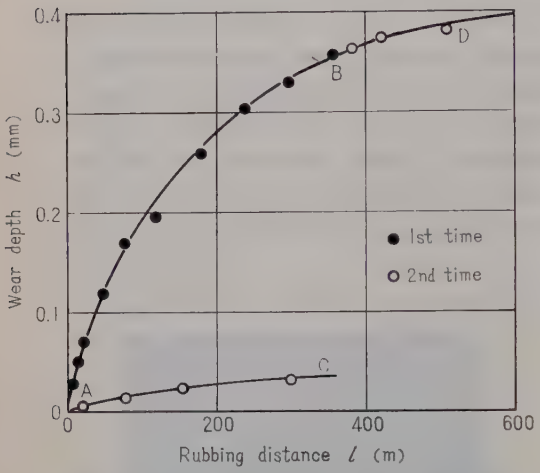

Fig. 9. Effect of repeated rubbing. Carbon steel SAE1040 and copper. Same conditions as those in Fig. 4.

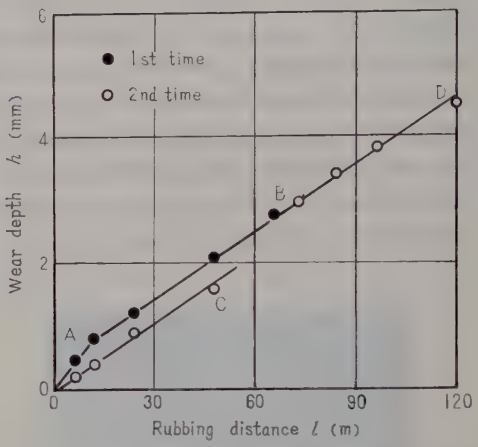

Fig. 10. Effect of repeated rubbing. A pair of specimens of carbon steel SAE1040. Same conditions as those in Fig. 5.

Similar results were obtained in a similar test with carbon steel against carbon steel in repeated rubbing (Fig. 10). As for the curves, AB is by the first test piece, AC by the second, and BD is the plot of AC as the extension of AB.

The results of the two tests depicted by Figs. 9 and 10 definitely confirm that, as had been inferred, accumulation of adhering worn particles on the rubbing surface is responsible for the transient state of wear in repeated rubbing.

Another test was carried out to examine the effect of adherence of worn particles. In this case, pipes of various diameters were prepared since the smaller the pipe, the more frequent the repetition of rubbing, and hence the greater should be the amount of accumulation in a given rubbing distance and the sooner the state of constant wear should be realized. In other words, the wear rate should become lower at a given rubbing distance if the diameter of the pipe is reduced. Fig. 11 shows the wear rates on a copper test piece by carbon steel (SAE1040) pipes of 29 mm and 69 mm average diameter, the number of revolutions being adjusted so that the rubbing velocity is equal in the two cases. The result is what was expected and confirms again that the accumulation of adhering worn particles has an essential influence upon the transition of the state of wear.

The tests described above disclose also that the work-hardened layer on the surface, if formed at all, does not seem to be concerned in the transition of the state of wear,

for, in the tests depicted in Figs. 9 and 10, the second test piece, freshly machined and devoid of any surface layer, takes over from the first—which had in all probability some hardened surface layer—with no deviation in the trend of wear rate variation. Again, in the test recorded in Fig. 11, despite the difference in pipe size and the as-

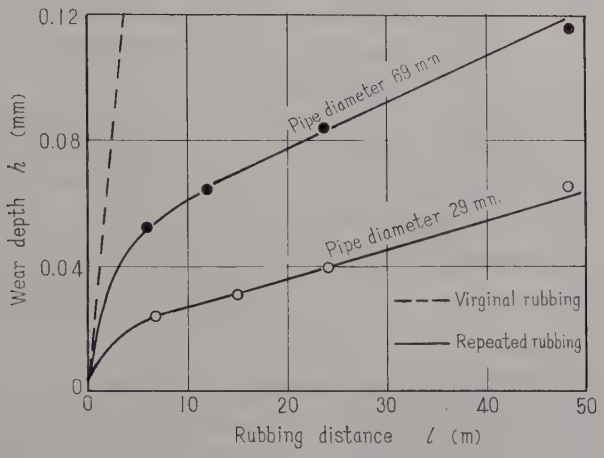

Fig. 11. Effect of numbers in repeated rubbing. Large test piece: carbon steel SAE1040. Small test piece: copper. Other conditions the same as those in Fig. 4.

sumption that the formation of the particular layer on the two test pieces at the end of equal rubbing distance proceeds at the same rate, the wear is less with the pipe of smaller size.

WEAR PROCESS UNDER LUBRICATION

Since almost all moving parts of machinery are lubricated, tests were made to ascertain whether the wear by friction under lubrication undergoes the same transition from initial to constant as has been observed in the dry state. Fig. 12 shows the result of tests on the wear of a copper test piece both by virgin friction and in repeated rubbing with continual dropping of machine oil onto the end face of a carbon steel pipe. On the figure, AB is by virgin friction, AC by repeated rubbing and CD is again by virgin friction with the cutting tool reset at C. All the characteristics in the trend of wear process seen in dry friction are faithfully reproduced: with virgin friction the wear is proportional to rubbing distance; in repeated rubbing the wear is much lighter than by virgin friction and the wear rate falls with the increase of rubbing distance; the layer formed on the surface and the varying roughness of the surface scarcely affect the wear. With carbon steel against carbon steel similar results are obtained, as shown in Fig. 13. Covering up the virgin surface and thus protecting it with minute worn particles has an important share in the "running-in" of moving parts of machinery to reduce the wear; formation of a work-hardened layer on the surface or

reduction of unevenness of the surface in order to lessen further wear is not the essential requirement for "running-in".

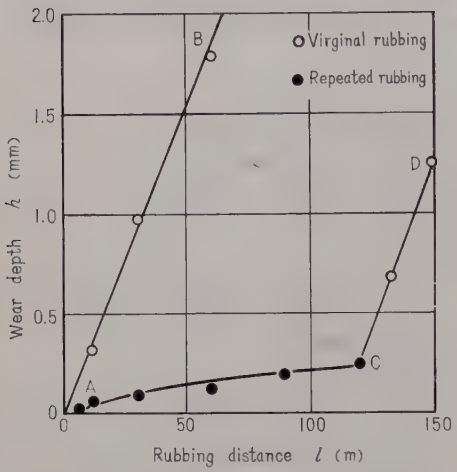

Fig. 12. Comparison between virginal and repeated rubbing under lubrication. Large test piece: carbon steel SAE1040. Small test piece: copper. Contact pressure: 78 kg/cm². Rubbing velocity: 7.5 cm/sec (machine oil lubrication).

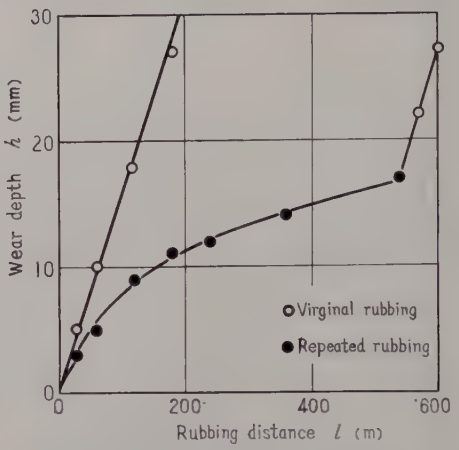

Fig. 13. Comparison between virginal and repeated rubbing under lubrication. A pair of specimens of carbon steel SAE1040. Other conditions the same as those in Fig. 12.

DISCUSSION

As has already been explained, the transient state of wear produced by friction with repeated rubbing either in the dry state or under lubrication is attributed not to some work-hardened layer formed on the rubbing surface nor to the surface roughness being varied but to gradual accumulation of adhering worn particles on the surface. These particles continue to accumulate, becoming broken up by load and friction or being oxidized by heat of friction, changing in shape and constitution till they drop out, thus establishing a constant condition of the surface layer, which in turn makes the wear rate also constant. The principal reason we have had for believing the growth of work-hardened surface layer to be responsible for the transient state of wear comes from the experimental evidence of the two processes progressing hand in hand. It is, however, more convincing to reason that the work-hardening is the result of the accumulation of adhering worn particles, for when a layer of such particles is formed, the rubbing is no longer by the naked pipe face, hence the wear rate would be reduced while the rubbing continues. This excess rubbing would affect the surface structure and work-harden it.

If the surface structure is affected by friction and the growth of a work-hardened layer improves the mechanical properties of the surface, the wear rate is bound to drop. In one of the previous investigations, when the wear on a surface, very much deformed and work-hardened by shot-peening, was compared with that on a surface electrolytically polished to remove every possible asperity, the former was at the

most 20–30% less than the latter[4]. Reduction in wear on a work-hardened surface by friction is estimated therefore as 20–30% at the most. Smoothening of the surface by repeated rubbing naturally reduces the wear rate; this reduction would be at the most in the order of several tens per cent, as can be seen from Fig. 7.

On the other hand, as shown in Fig. 4, the constant wear rate of copper when rubbed repeatedly against carbon steel falls to as low as 1/10,000 of its initial wear rate, which proves how great is the effect of the layer of adhering minute worn particles in reducing the wear. With certain combinations of materials or under certain rubbing conditions, cases are known in which the constant wear surpasses the initial, probably because the worn particles are scratching the surface, producing the lapping effect and thus quickening the wear. Depending upon the rubbing velocity, contact pressure and other factors, worn particles are known to assume different states: as original metal, as oxidized and as fused. Accordingly, the effect they produce upon wear will be different; it can be protecting, it can be lapping. The problem of the how and why of these effects is being left for future study.

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REIBUNG UND SCHMIERUNG IM MASCHINENBAU, INSBESONDERE BEI ZUSÄTZLICHER VERWENDUNG VON KOLLOIDAL-GRAPHIT*

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SUMMARY

FRICTION AND LUBRICATION IN ENGINES, IN PARTICULAR WITH ADDED COLLOIDAL GRAPHITE

The corrosive action of HP additives and moisture on the surface of cast iron cylinder walls and the vulnerability of the Beilby layer are discussed. Protection of surfaces by colloidal graphite has been studied in laboratory and field tests. HP-additives do not interfere with the beneficial effects of graphite. Graphitized cylinder walls of an Otto-motor were exposed to unfavourable climatic conditions in the laboratory, corrosion resistance was higher than that of a motor protected only by an anti-corrosion oil.

Der an Gleitflächen auftretende Verschleiß beruht bekanntlich nicht ausschließlich auf mechanischem Abrieb, sondern es können chemische und vor allem elektrochemische Korrosionen infolge der Bildung von Lokalelementen auftreten, die sich

Abb. 1. Korrosions-Standversuch im Ölbad. Werkstoff: GG-Zylinderlaufbüchsen.

auf den Gesamtverschleiß u.U. beachtlich auswirken können. Bei der Bildung solcher Lokalelemente wirkt das Schmieröl als Elektrolyt.

* Vortrag gehalten am 13. Mai 1959 auf dem Kolloquium der Deutschen Rheologischen Gesellschaft, Berlin-Dahlem.

In Abb. 1 werden 6 Streifen aus einer GG-Zylinderlaufbüchse gezeigt, die 150 Tage lang in je einem offenen, mit handelsüblichen Schmierölen gefüllten Becherglas gestanden haben. Die 3 ersten dieser Gläser waren mit einem Premiumöl, die 3 restlichen mit einem HD-Motorenöl gefüllt worden. Um den Einfluß der Feuchtigkeit im Schmieröl kennen zu lernen, waren dem Öl in den Gläsern 2 und 5 je 1% H_2O in den Gläsern 3 und 6 je 5% H_2O zugesetzt worden. Man sieht schon aus den nicht vergrößerten Aufnahmen die auffallenden Unterschiede im Aussehen der Versuchsstreifen nach Abschluß des Standversuches. Deutlicher wird die Auswirkung der elektrochemischen Korrosion durch den Vergleich der Abb. 2 und Abb. 3, welche die

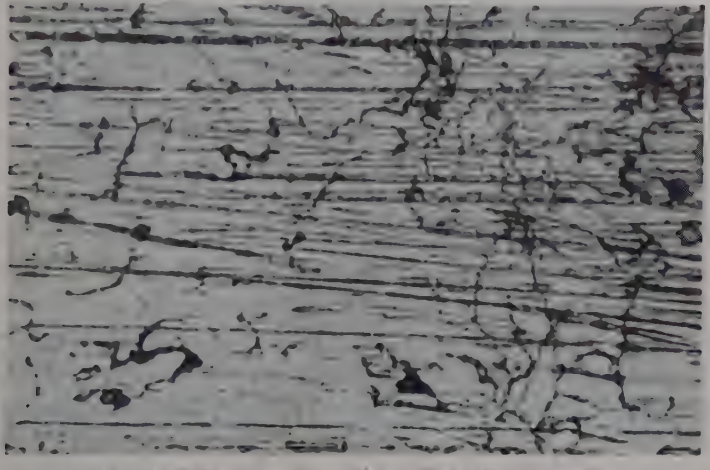

Abb. 2. Probe 1 nach 150 Tagen Standzeit (× 150).

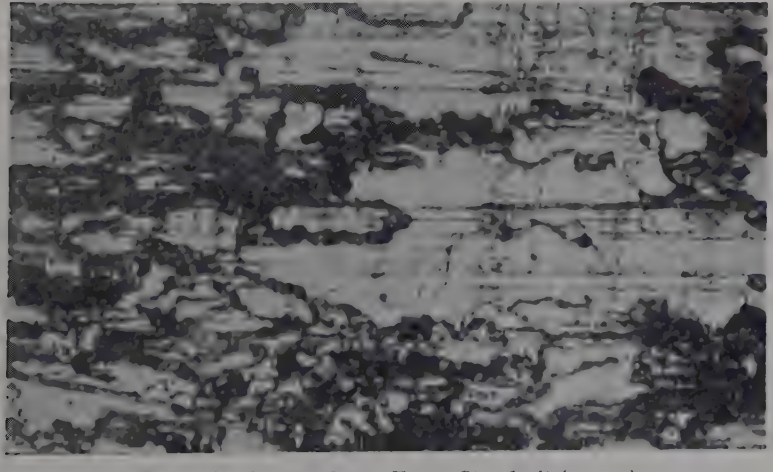

Abb. 3. Probe 4 nach 150 Tagen Standzeit (× 150).

Oberflächen der Streifen 1 und 4 in 150-facher Vergrößerung zeigen. Während man bei dem mit schwach legiertem Öl benetzten Streifen 1 in Abb. 2 außer den Bearbeitungsriefen keinerlei Beschädigung an der Oberfläche feststellen kann, zeigen sich an der in

HD-Öl gestandenen Probe 4 starke Anfressungen. Mit Hilfe einer als Ersatzelement dienenden Meßzelle und entsprechender Meßinstrumente gelang es, die Wirkung und die Veränderung der elektrolytischen Eigenschaften von Schmierölen auch im praktischen Motorbetrieb mit ausreichender Genauigkeit zu bestimmen[1].

Da im Laufe dieser Versuche festgestellt wurde, daß bei Verwendung von HD-Ölen weder eine Ablagerung von Ölschlamm, noch ein Festsetzen von Kolbenringen usw. eintraten, diese Öle also grundlegende Vorteile gegenüber nicht- oder schwachlegierten Schmierölen aufweisen, wurde versucht, unter Beibehaltung der Vorteile der HD-Öle deren Nachteil, die elektro-chemische Korrosion, zu beheben. Ausgangpunkt dieser Überlegungen war die sogenannte *Beilby-Schicht*. Auch sorgfältigst bearbeitete Laufflächen bestehen, in entsprechender Vergrößerung dargestellt, aus nebeneinander liegenden „Gebirgsketten und Tälern", deren Richtung durch den Weg des spanabhebenden Werkzeuges bedingt ist. Elektronenmikroskopische Untersuchungen ergeben, daß die Kämme der Hügelketten kristallin sind und dadurch die Bildung von Lokalelementen und deren Auswirkungen ermöglichen. Es ist klar, daß bei den ersten Gleitbewegungen von zwei Oberflächen vor allem ein rein mechanischer Abrieb der Unebenheiten eintritt, daß mit zunehmender Glättung der Gleitflächen der Verschleiß abnimmt, wobei nach Ansicht BEILBY's der Lokaldruck an den tragenden Berührungsflächen der Hügel auch bei homogenem Ölfilm immer noch so hoch ist, daß an den Druckstellen eine plastische Verformung des Materials eintritt und dieses in das benachbarte Tal abfließt. Dabei sinkt die Druckhöhe sehr rasch ab und der Werkstoff erstarrt, ehe eine Kristallbildung eintreten kann. Die Folge davon ist, daß die Oberfläche amorph wird. Damit entfällt die Möglichkeit der Bildung von Lokalelementen, die Schicht wird ferner sehr widerstandsfest gegen mechanischen Abrieb und weist außerdem recht beachtenswerte Gleiteigenschaften auf.

Leider gibt es bis heute keine Möglichkeit, in der Praxis durch Steuerung der Vorgänge während der Einlaufperiode Beilby-Schichten zu erzeugen. Ebenso kann man mit Hilfe der heutigen Diffraktionstechnik noch nicht einwandfrei feststellen, ob es sich bei den Beilby-Schichten wirklich um rein amorphe oder mikrokristalline Schichten handelt. Die Arbeiten von HOPKINS, FINCH UND GLOCKLER unterstützen die Auffassung, daß es sich um rein amorphe Schichten von etwa 20 Å Dicke handelt[2]. RAETHER dagegen vertritt den Standpunkt, daß es ebenso gut eine mikrokristalline Schicht sein kann[3]. Diese ist gegen elektrochemische Korrosionsangriffe ebenfalls weitgehend unempfindlich. Da auch kleinstkristalline Schichten sich nicht nach Wunsch erzeugen lassen, wurde in Erwägung gezogen, eine *monokristalline Schicht* zu schaffen und hierbei lag es nahe, auf Grund ihrer Kristallstruktur Graphit und MoS_2 auf ihre Eignung zu untersuchen.

Man weiß, daß zwei glatte Oberflächen mit Rauhigkeiten innerhalb der Grenzen von Atom- oder Molekulargrößen, auch wenn sie gegeneinander gepreßt werden, nicht unmittelbar aufeinander liegen, sondern durch sehr dünne, aber trotzdem wirksame adsorbierte Zwischenschichten voneinander getrennt sind. Diese spielen vor allem im Bereich der Grenzschmierung eine wichtige Rolle[4]. Abb. 4 zeigt in schematischer Darstellung die Schichten zweier geschmierter Metalloberflächen.

Wenn mit 1 das Werkstück selbst bezeichnet wird, so stellt 2 die *Tragschicht* dar, deren Oberflächenbeschaffenheit, insbesondere deren Rauhigkeit, durch die Art der Bearbeitung bedingt ist[5].

Die darüber liegende Oxydschicht entsteht durch die Oxydation der Oberfläche frisch bearbeiteter Werkstücke und erreicht normalerweise eine Schichtdicke bis zu

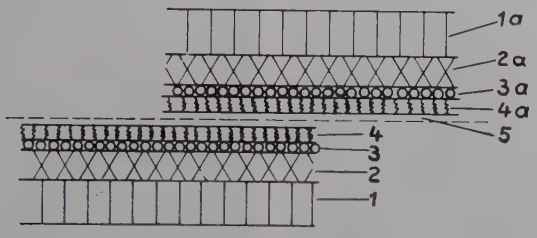

Abb. 4. Schema eines Gleitelementes.
1 u. 1a Metallstütze, 2 u. 2a Tragschicht, 3 u. 3a Oxydschicht, 4 u. 4a Grenzschicht, 5 Ölfilm.

100 Å und mehr. Oxydschichten sind praktisch immer amorph, haften fest an ihrer Unterlage und sind isotropisch, d.h. sie haben nach allen Seiten hin die gleiche Festigkeit. Bei weicher Tragschicht und hohen Gleitgeschwindigkeiten neigen Oxydschichten leicht zum Zerreissen und ermöglichen dann Kaltverschweissungen. Durch Zugabe sogenannter Hochdruck-Additive kann man diese Schichten jedoch chemisch verändern und dabei ihre Festigkeit wesentlich erhöhen.

Ein weiterer Nachteil der Oxydschichten besteht in den hohen Reibungswerten, die man durch Aufbringen einer weiteren Schicht, der *Grenzschicht*, zu senken versucht. Diese letztere ist zwischen 50 und 200 Å dick und besteht aus Schmierstoffmolekülen, die sich in unmittelbarer Nähe der Oxydschicht gegenüber dieser nach einem bestimmten System orientieren und mit zunehmendem Abstand in den unorientierten flüssigen Zustand übergehen[6].

Es sollte nun versucht werden, auf die Oxydschicht eine Zwischenschicht aus den obenerwähnten festen Schmierstoffen Graphit oder MoS_2 aufzubringen. Die Kristallstruktur der beiden Stoffe ist bekannt, ebenso die guten Reibungseigenschaften, insbesondere in Verbindung mit einem flüssigen Schmiermittel. Es handelt sich bei beiden um anisotropische Körper mit Schichtstrukturen, die auch höheren senkrechten Drücken widerstehen können, deren Scherfestigkeit gegen Kräfte in Richtung der Schichtebenen dagegen gering ist. Nach Untersuchungen von SAVAGE[7] hat es den Anschein, als ob die geringe Reibung und der niedrige Verschleiß, durch die sich Graphitoberflächen normalerweize auszeichnen, ebenfalls auf dem Vorhandensein adsorbierter Schichten beruhen. Diese bilden sich unter dem Einfluß der atmosphärischen Feuchtigkeit von selbst, ebenso erfolgt bei Verletzung der Schicht eine Neubildung automatisch.

Die hohe Adsorptionsfähigkeit des Graphits[6] erleichtert das Auf- bzw. Einbringen entsprechender Deckschichten auf bzw. in Gleitflächen. In den Abb. 5 und 6 sind je

zwei entsprechende Mikroaufnahmen von Zylinderlaufbahnen aus 4-Zylinder-Ottomotoren in rd. 1000-facher Vergrößerung gegenübergestellt. Mit beiden Motoren war unter vollkommen gleichen Bedingungen ein 100-Stundenlauf durchgeführt worden. Beide Motoren hatten das gleiche HD-Öl, doch wurde im ersteren Falle dem

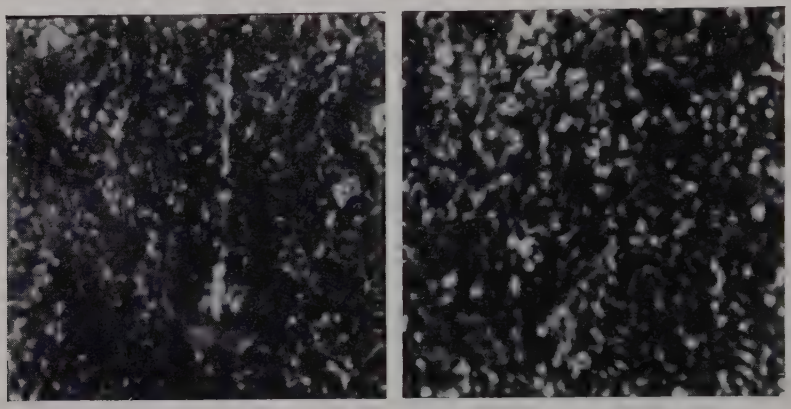

Abb. 5. Mikroaufnahme einer Laufbüchse nach 100 h Lauf (× 100). Schmieröl mit Graphitzusatz.

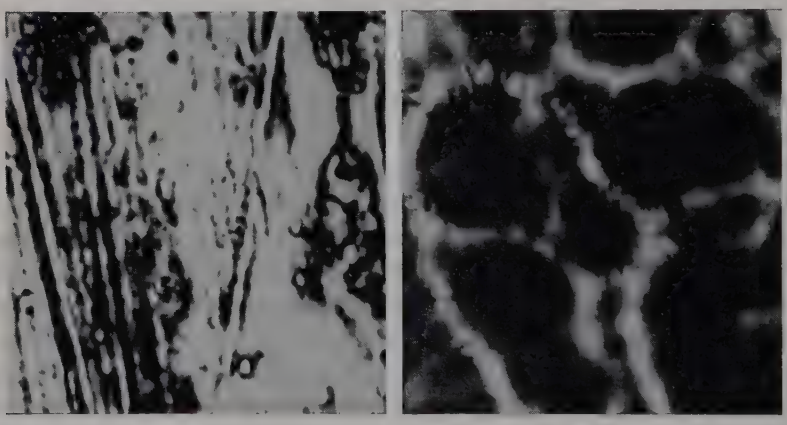

Abb. 6. Mikroaufnahme einer Laufbüchse nach 100 h Lauf (× 100). Schmieröl ohne Graphitzusatz.

Öl 1.5% Autokollag beigemischt. Man erkennt in Abb. 5 die praktisch homogene Graphitschicht, die sich aus dem Schmieröl an die Zylinderwand niedergeschlagen hat und durch den von den Kolbenringen ausgeübten Normaldruck in den Zylinderwerkstoff „eingeplättet" worden ist. In der linken Darstellung der Abb. 6 erkennt man noch genau die Bearbeitungsriefen, in der rechten ist ein Ansatz von Rost erkennbar. Die vier Aufnahmen stellen Ausschnitte aus dem obersten Teil der Laufbüchsen dar, wo durch die hohe thermische Belastung der Schmierölfilm geschwächt wird und dadurch bei Auftreten von Verbrennungswasser und unter dem Einfluß der übrigen Verbrennungsprodukte Korrosionen an der Zylinderwandung leicht möglich sind. Bei einer Nachprüfung im Fritz-Haber-Institut wurde an Hand elektronenmikros-

kopischer Aufnahmen, Elektronenbeugungs- und Einkristalldiagrammen bestätigt, daß eine Einlagerung hexagonaler Graphitplättchen in die Gleitflächen der Laufbüchsen des ersten Motors stattgefunden hatte[8].

In weiteren Versuchsreihen wurde untersucht, aus welchem Grunde bei den obenangeführten Dauerläufen der Graphitniederschlag auf dem Grauguß-Zylinder und nicht auf dem Leichtmetall-Kolben erfolgt war. Zu diesem Zweck wurden unter Verwendung von Elektrodenplatten aus LM-Kolben- und GG-Zylindermaterial elektrolytische Zellen zusammengebaut und an 220 V-Gleichstrom angeschlossen. Als Elektrolyt dienten handelsübliche legierte und unlegierte Schmieröle sowie Kraftstoffe, denen allen koloidaler Graphit oder ebensolches MoS_2 in den von deren Herstellerfirmen angegebenen Mengen zugemischt wurde. Hierbei ergab sich folgendes: Bei mehr als 150 Versuchen unter allen möglichen Variationen der Versuchsbedingungen setzte sich in jedem Falle ein homogener Niederschlag der Kolloide an einer Elektrodengruppe ab, ein Beweis dafür, daß die Anwesenheit von HD-Zusätzen die Wirkung der kolloidalen festen Schmierstoffe weder verhindert noch einschränkt. Auffallend war jedoch die Tatsache, daß, je nach der Art des Elektrolyten, die Niederschläge sich an den Leichtmetall-Elektroden bildeten — ohne Rücksicht darauf, ob diese positiv oder negativ waren — oder an den Plus-Elektroden, gleichgültig, ob der Werkstoff LM oder GG war. In einem einzigen Fall wurde die Graphitschicht an 2 negativen Elektroden festgestellt, von denen die eine ebenfalls aus LM, die andere aus GG bestand. Durch zusätzliche Teste wurde gefunden, daß die pH-Werte der Elektrolyten hinreichend genaue Voraussagen zulassen, an welche Elektrodengruppe sich die Kolloide niederschlagen. Der Einfluß jedoch, den die einzelnen Bestandteile bzw. Zusätze der Elektrolyten auf die Polarität von Graphit und MoS_2 ausüben, konnte noch nicht ermittelt werden.

Soll die natürliche Bildung einer Graphitschicht aus einem graphitierten Schmieröl oder Kraftstoff heraus, die eine gewisse Zeit erfordert, nicht abgewartet werden, sondern von vornherein eine Reibung und Verschleiß mindernde sowie gegen Korrosion schützende Deckschicht auf eine Oberfläche aufgebracht werden, so kann man verschiedene Wege gehen:

(a) Man kann aus dem kolloidalen festen Schmierstoff und einer geeigneten Trägersubstanz eine Dispersion herstellen und diese durch Aufstreichen, Aufspritzen oder Einreiben auf die gewünschte Fläche aufbringen. Als Nachteil dieses Verfahrens kann die verhältnismäßig geringe Haftung angeführt werden, infolge derer ein großer Teil der aufgebrachten Schicht zu Beginn des Einlaufens wieder abgerieben und vom Schmieröl abgeführt wird.

(b) Man kann die Kolloide in einer z.T. aus Kunstharz bestehenden Trägersubstanz in der Schwebe halten und ebenfalls durch Streichen oder Spritzen aufbringen. Um die Haftung noch zu verbessern, wird in der Regel die zu schützende Fläche vorher durch Sandstrahlen oder Ätzen aufgerauht. Die so geschaffene Schutzschicht wird noch fester und dauerhafter, wenn man Harze verwendet, die nach dem Aufbringen durch Einbrennen gehärtet werden können[9].

(c) Soweit es sich um Graphit handelt, kann man die Deckschicht ohne Mühe

galvanisch erzeugen. Dies kann auf verschiedene, zum großen Teil in der Literatur veröffentliche Verfahren erfolgen[8].

Daß eine homogene Deckschicht aus festen Schmierstoffen beachtliche Notlaufeigenschaften aufweist und einen wirksamen Schutz gegen Korrosionsangriffe darstellt, mag durch folgende Versuchsergebnisse dargelegt werden:

Ein luftgekühlter Einzylinder-Motor, auf dessen Zylinderlauffläche eine aushärtbare Graphit-Kunstharzdispersion aufgespritzt worden war, wurde mit einem Elektromotor gekuppelt und lief auf dem Prüfstand 8 Stunden lang mit Fremdantrieb bei 3000 U/min ein. Die Benzinzufuhr war abgestellt, die Zündung dagegen eingeschaltet. Als Schmieröl wurde ein unlegiertes Markenöl verwendet. Als nach den 8 Stunden das Öl abgelassen worden war, lief der Motor 6 weitere Stunden under den gleichen Bedingungen, ohne daß sich das geringste Anzeichen einer Störung zeigte. Darauf wurde der Zylinder abgenommen und die Zylinderlaufbahn und der Kolben mit Benzin gewaschen. Nach dem Wiederzusammenbau lief der Motor mit „trockenem Zylinder" unter den gleichen Bedingungen wie oben noch 2 Stunden 47 Minuten. Nach dieser Zeit trat das bekannte Kolbenklemmen ein, ohne daß allerdings ein Schaden am Motor entstanden wäre. Bei einem nicht vorgraphitierten Motor trat dieses Klemmen infolge der höheren Reibung und der dadurch bedingten größeren Reibungswärme 3 Stunden 45 Minuten nach Ablassen des Schmieröles ein.

Zu diesen Ergebnissen ist folgendes zu bemerken: Die angegebenen langen Laufzeiten sind nur dadurch erreicht worden, daß der Motor fremd angetrieben wurde. Denn als bei einem späteren Versuch mit dem gleichen Motor nach 8-stündigen Einlauf das Öl abgelassen und die Maschine mit eigener Kraft bei 2/5 der Vollast und 3000 U/min weiterbetrieben wurde, trat nach *ca.* 45 Minuten ein Fressen der Kolben ein, gleichgültig, ob die Zylinder vorgraphitiert worden waren oder nicht. Bei graphitierten Zylindern jedoch machen sich die Notlaufeigenschaften der Deckschicht insofern bemerkbar, als das Kolbenklemmen allmählich eintritt und am langsamen Abfall der Drehzahl erkannt wird, sodaß durch Abstellen des Motors Schaden verhütet werden kann. Der Grund für das Festgehen der Kolben in dem letzten Versuch ist darin zu sehen, daß beim Fehlen des Ölumlaufs die in jedem Arbeitshub entstehende Wärme nicht genügend abgeführt wird. Infolgedessen wird der LM-Kolben thermisch sehr hoch belastet und dehnt sich infolge der Unterschiede der Ausdehnungskoeffizienten erheblich mehr aus als der GG-Zylinder.

Als Beweis für die größere Widerstandsfähigkeit graphitierter Flächen gegenüber der nicht graphitierter möchte ich das Ergebnis der Untersuchung von 2 Zylindern wassergekühlter Otto-PKW-Motoren im Fritz-Haber-Institut anführen:

1. Der Zylinder des Motors a, der mehr als 120.000 km mit einem unlegierten, aber graphitierten Schmieröl gefahren wurde, ist gegen Ätzangriffe indifferenter als der Zylinder des Motors b, der 86.000 km mit einem Marken-HD-Öl gelaufen war.

2. Die Lauffläche des Zylinders aus Motor a hat sich in molekularen Größen verändert und ist dadurch widerstandsfester gegen chemische und elektrochemische Korrosion und damit gegen mechanischen Abrieb geworden. (Das letztere ergab sich auch aus den Verschleißwerten).

Analog sind Resultate aus Versuchen im eigenen Institut. Es sollte festgestellt werden, ob durch ein einfach anzuwendendes Mittel das Innere eines Motorzylinders wirksam gegen Rost geschützt werden kann, selbst wenn der Motor unter ungünstigen klimatischen Bedingungen gelagert wird. Es wurden zu diesem Zweck Zweitakt-Motoren 30 Minuten lang mit einem normalen Zweitaktgemisch 1 : 25 gefahren, anschließend 5 Minuten lang mit genau dem gleichen Gemisch unter Zusatz von 5% eines Korrosions-Schutzöles. Beim Stand des Kolbens im unteren Totpunkt wurde anschließend der Zylinder abgehoben, 2 Stunden lang in einem Kühlschrank bei einer Temperatur von —2 bis —5°C aufbewahrt und dann bei Raumtemperatur senkrecht aufgestellt. Bei sämtlichen „Leerversuchen", d.h. Versuchsläufen, bei denen der 5-Minutenlauf unter Verwendung von Kraftstoff plus Korrosions-Schutzöl wegfiel, sowie bei den meisten Läufen mit Schutzöl trat ein mehr oder weniger starker Rostansatz auf. Bei einem galvanisch graphitierten Zylinder dagegen wurde der Versuch in der Form des „Leerversuches" 8 Mal hintereinander durchgeführt, ohne daß sich ein Anflug von Rost gezeigt hätte. Erst beim 9. Lauf wurde in der Nähe des Auspuffschlitzes leichter Rostansatz festgestellt. Das Ergebnis der elektronenmikroskopischen Untersuchungen der Zylinderoberfläche steht noch aus.

Die in den obigen Ausführungen skizzierten Arbeiten werden laufend weitergeführt und sind inzwischen auch auf MoS_2 ausgedehnt worden.

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ZUR FRAGE DER ERMÜDUNGSERSCHEINUNGEN BEI WÄLZLAGERN

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ZUSAMMENFASSUNG

In dem vorliegenden Aufsatz werden die bisher veröffentlichten Untersuchungen zur Pitting-Bildung bei Wälzlagern und die daraus gewonnenen Hypothesen über die Ausfallursachen zusammengefasst, um einen allgemeinen Überblick über das ganze Problem zu erhalten. Dadurch könnte eine gewisse Klärung dieser Fragen herbeigeführt und Anregungen für weitere Versuche gegeben werden.

Nach einer Abgrenzung der Pittings als Festigkeitserscheinung gegenüber Verschleißerscheinungen werden die Versuchsergebnisse in folgende vier Abschnitte gegliedert: (1) Einfluß der Behandlung und Bearbeitung des Werkstoffes; (2) Spannungsverteilung unter dem Einfluß äußerer Belastungen; (3) Einwirkung des Schmiermittels; (4) Metallurgische Erscheinungen.

In den Schlußfolgerungen werden dann zwei mögliche Methoden angegeben, eine Lebensdauertheorie der Wälzlager aufzubauen. Die erste geht aus von empirischen Daten, die in entsprechender Weise verallgemeinert werden müssen. Dazu sind jedoch umfangreiche Versuche zur Ermittlung der einzelnen Einflüsse notwendig, um zu hinreichend genauen Ergebnissen zu kommen. Die zweite geht aus von Erkenntnissen über den Ermüdungsvorgang und seine Ursache. Hierzu muß von metallurgischer Seite her die Basis für weitere Untersuchungen geschaffen werden.

SUMMARY

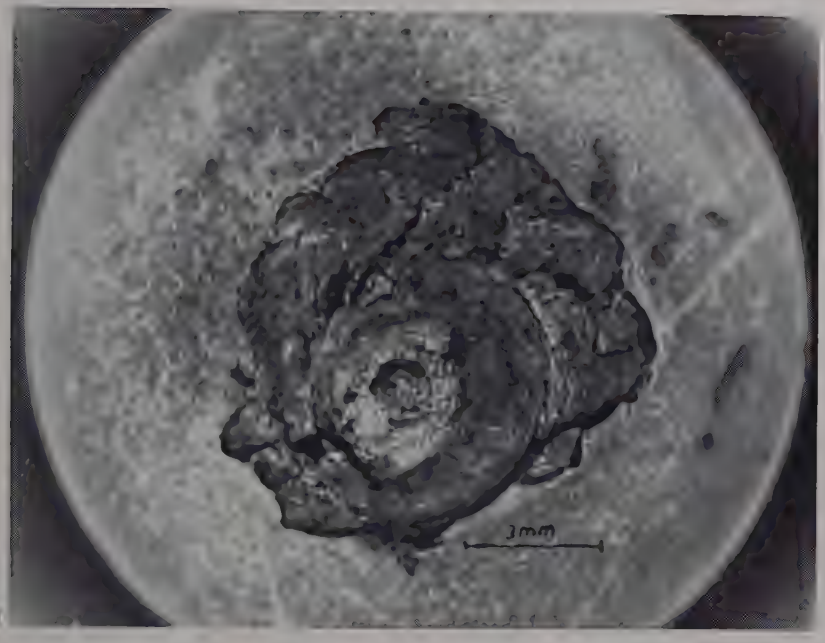
A review is given of investigations to date on pitting of antifriction bearings and of resulting hypotheses in order to obtain what might be called a general view on the whole problem. A discussion would help to clarify the situation and furnish new aspects for further experimental work.

The study deals first with pitting as a phenomenon of strength in contrast to phenomena of wear. Experimental results are then individually grouped under the following headings: (1) effects of material treatment and processing; (2) stress distribution caused by external loads; (3) effect of lubricant; (4) metallurgical phenomena.

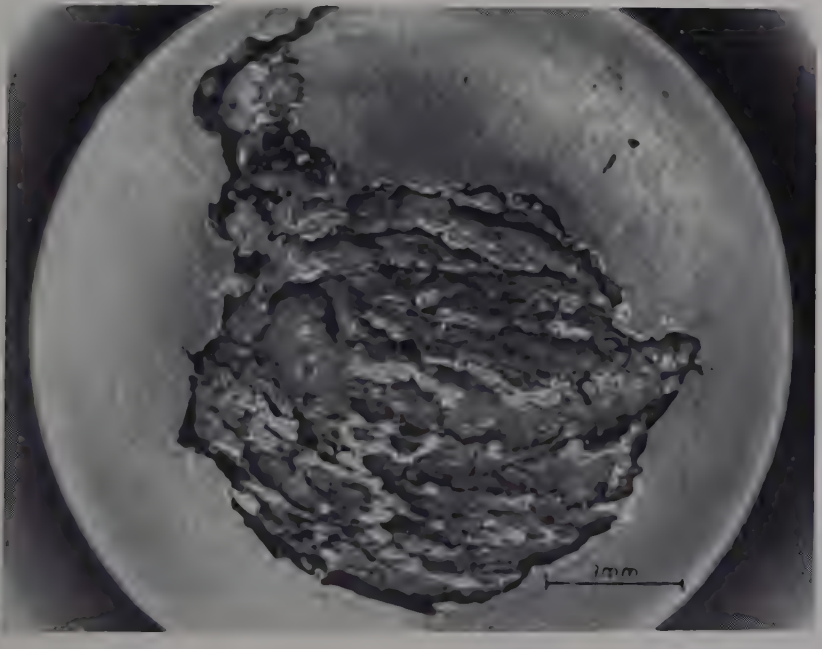
Two methods for devising a life theory for antifriction bearings are presented, the first based on empirical data. Extensive investigations of the various influences will still be required, however, to arrive at sufficiently exact statements. The starting point for the second method is the knowledge about the process of fatigue and its underlying causes. Here, metallurgy will have to work out a basis for further fatigue life studies.

Bei Wälzlagern tritt erfahrungsgemäß nach längerem Lauf eine Ausfallerscheinung auf, die als Grübchen- oder Pittingbildung bezeichnet wird. Anfänglich sehr kleine Risse oder porenähnliche Vertiefungen in den Oberflächen der aufeinander abrollenden Körper weiten sich im Laufe weiterer Überrollungen sehr schnell zu größeren,

meist schollenartigen Ausbrüchen aus. Oft treten solche Schäden auf, ohne daß vorher, auch bei sorgfältiger Untersuchung der Oberfläche, Anzeichen für eine beginnende Grübchenbildung bemerkt werden. Das Aussehen solcher Grübchen ist aber



(a)



(b)

(c)

Abb. 1. (a) (b) und (c) Typische Pittingbildungen.

bei allen Wälzlagern sehr ähnlich, vorausgesetzt, daß eine solche Ausbröckelung nicht noch zu oft überrollt und dadurch das Schadensbild verändert wurde. In Abb. 1 sind einige typische Pittings dargestellt. Man erkennt daran recht deutlich, daß eine Oberflächenschicht von annähernd konstanter Dicke ausgebröckelt ist. Da diese Pittingbildung die Lebensdauer eines Wälzlagers beendet, hat man sich schon in den zwanziger Jahren damit befaßt, diese Erscheinungen näher zu untersuchen, um eine Basis für die Berechnung der Wälzlager zu finden.

Man war sich damals aber bald darüber im klaren, daß alle Deutungsversuche nicht dazu geeignet waren, einen mathematischen Zusammenhang zwischen den einzelnen Einflußgrößen aufzustellen. Man versuchte daher, auf empirischem Wege über umfangreiche Lebensdauerversuche mit Wälzlagern zu einer Berechnungsgrundlage zu kommen. Daraus resultierte die heute in Deutschland nach DIN 622 genormte Beziehung, nach der die Lebensdauer eines Lagers umgekehrt proportional ist der dritten Potenz der Lagerbelastung. Auf dieser Basis konnte nun zwar der ungefähre Zeitpunkt des Lagerausfalles abgeschätzt werden, eine Erklärung bzw. wesensmäßige Erfassung der Ausfallerscheinungen wurde damit aber nicht gegeben. Im Laufe der Zeit wurden nun zahlreiche Untersuchungen durchgeführt, die Erkenntnisse über den Ausfallvorgang und den Einfluß einzelner Faktoren brachten. Oft wurden dabei Hypothesen über die möglichen Ursachen aufgestellt, wobei in vielen Fällen Einzelheiten überbewertet wurden und somit ein einseitiges Bild der zum Ausfall führenden Vorgänge gewonnen wurde.

Im folgenden soll ein Überblick über die verschiedenen Ergebnisse und die daraus gewonnenen Ausfallhypothesen, soweit sie veröffentlicht worden sind, gegeben werden. Im letzten Jahr ist von ERDMANN-JESNITZER UND WEIGEL[18] im Rahmen von Untersuchungen zur Pittingbildung an ungehärteten Zahnrädern ein Abriß über den

Stand der Forschung auf diesem Gebiet gegeben worden. Manche der dort zitierten Arbeiten und Ergebnisse werden auch in dem vorliegenden Aufsatz besprochen werden, wobei aber hier der Themastellung gemäß die Vorgänge bei gehärteten Werkstoffen im Vordergrund stehen.

Um in das vorliegende Material eine gewisse Ordnung zu bringen, werden folgende Punkte betrachtet, die für die Ausfallerscheinungen wesentlich sind, nämlich:

Behandlung und Bearbeitung des Werkstoffes

Spannungsverteilung unter dem Einfluß äußerer Belastungen

Einwirkung des Schmiermittels

Metallurgische Erscheinungen.

Bevor auf die einzelnen Punkte eingegangen wird, soll zuerst das zu behandelnde Gebiet abgegrenzt werden.

VERSCHLEISS- ODER FESTIGKEITSERSCHEINUNG?

Diese Frage war eine der ersten, die bei der Untersuchung der Wälzlagerausfälle gestellt wurden. Hierbei soll unter Verschleiß sowohl mechanischer Abrieb als auch chemischer Angriff, wie Korrosion, Absorption, Reiboxydation usw. verstanden werden.

Es wurden viele Versuche und Untersuchungen mit Wälzlagern durchgeführt, um zu entscheiden, wie groß der mechanische Abrieb während des Laufes ist und ob dieser Verschleiß ausschlaggebend für die Lebensdauer des Lagers ist. Dabei hat sich herausgestellt, daß der Abrieb einwandfrei eingebauter und geschmierter Lager im normalen Lebensdauerversuch so gering ist, daß er als Ausfallursache nicht in Frage kommt. Wenn Verschleiß in großem Maße auftrat, konnte immer nachgewiesen werden, daß er auf außergewöhnliche Ursachen zurückzuführen war, wie z.B. Staub und andere schmirgelnde Stoffe, Blockieren des Lagers usw.

CORDIANO u.a.[13] haben in letzter Zeit Untersuchungen durchgeführt und dabei gefunden, daß Korrosionserscheinungen nicht als Ausfallursache anzusehen sind. TRUBIN[70] machte die gleichen Erfahrungen.

KARPENKO (unter[73]) spricht von einer Absorptionsermüdung, die durch inaktive Stoffe hervorgerufen wird. Diese sollen an ihrer Oberfläche aktive Stoffe ansammeln, die ihrerseits dann die Metalloberfläche angreifen. Eine derartige Substanz sei z.B. Oleïnsäure. Er hat festgestellt, daß durch diese Erscheinungen die Lebensdauer um 10–20% erniedrigt wird. FINK UND HOFMANN (unter[38,72]) nehmen an, daß für die Ausfallerscheinungen an überrollten Metalloberflächen Reiboxydation verantwortlich ist, die im Laufe der Zeit die Oberfläche porös macht und so zu Ausbröckelungen führt. Diese Reibrostbildung trete nur auf in Anwesenheit von Luftsauerstoff oder von Sauerstoff, der im Schmiermittel gelöst ist. DIERGARTEN[16] spricht in analoger Erweiterung nicht nur von einer Reiboxydation, sondern auch von einer Hydridation.

LANCHESTER (unter[38,72]) hat eine andere Verschleißart zur Erklärung von Ausfällen an überrollten Metalloberflächen herangezogen. Er geht davon aus, daß kleine Metallpartikel infolge der durch die Reibungseffekte auftretenden Temperaturspitzen angeschweißt und bei den darauf folgenden Überrollungen wieder abgerissen werden.

Dadurch soll eine Oberflächenverletzung entstehen, die sich im Laufe der Zeit in einer Grübchenbildung bemerkbar macht. Auch bei TRUBIN[70] ist diese Deutungsart angeführt, jedoch mit der Bemerkung, daß in der Praxis derartige Erscheinungen nicht bestätigt werden konnten.

Aus eigenen Erfahrungen kann ergänzend dazu gesagt werden, daß Untersuchungen an gelaufenen Wälzlagern keine Anzeichen für einen derartigen Verschleißvorgang gebracht haben. Es konnten zwar manchmal eingewalzte Metallteilchen festgestellt werden, die aber in jedem Fall entweder von schon vorhandenen Grübchen stammen oder aber mit dem Schmiermittel von außen herangeführt wurden.

Bei der Betrachtung der verschiedenen Verschleißerscheinungen kann festgestellt werden, daß keine als Ursache für den Ausfall im Normalfall in Frage kommt, wobei unter Normalfall, wie schon oben erwähnt, ein einwandfreier Lauf ohne äußere Einwirkungen verstanden werden soll. Vielmehr deutet die Gleichartigkeit der entstehenden Grübchen auch unter verschiedensten Bedingungen sowie gewisse Gesetzmäßigkeiten zwischen Beanspruchung und Laufzeit darauf hin, daß es sich um eine Ermüdungserscheinung des Werkstoffes handelt. Der Werkstoff, in diesem Zusammenhang handelt es sich ausschließlich um Stahl, hält hierbei für eine gewisse Zeit ein Überrollen aus, um dann bei längerer Beanspruchung unter Pittingbildung auszufallen. Damit liegen die Ermüdungsausfälle bei Wälzlagern auf derselben Ebene wie die Zeitfestigkeit bei anderen wechselbeanspruchten Bauteilen, ohne allerdings, wie weiter unten gezeigt wird, mit diesen direkt vergleichbar zu sein. Wenn auch gewisse wesensmäßige Zusammenhänge hinsichtlich der Schadensursachen bestehen, handelt es sich doch auf Grund der vollkommen andersgearteten Beanspruchungsweise um eine andere Festigkeitseigenschaft des Werkstoffes, für die, wie nun gezeigt werden soll, die verschiedensten Einflüsse entscheidend sind, so daß es sich hierbei um ein ziemlich komplexes Verhalten des Werkstoffes handelt im Gegensatz zu der üblichen Wechselfestigkeit.

EINFLUSS DER BEHANDLUNG UND BEARBEITUNG DES WERKSTOFFES

Eine der zuerst aufgestellten Hypothesen über die Ursache der Ermüdung von Wälzlagern geht von den im Stahl vorhandenen Inhomogenitäten aus. Als solche Störungen im Werkstoff sind einmal Schlackeneinschlüsse jeder Art und zum anderen aus der Warmbehandlung resultierende Unregelmäßigkeiten wie Karbidanhäufungen, Strukturunregelmäßigkeiten, unterschiedliche Gefügeausbildungen usw. zu verstehen. Man geht dabei von der Vorstellung aus, daß aus den Inhomogenitäten örtliche Spannungserhöhungen erwachsen. Die durch die einzelnen Belastungszyklen hervorgerufenen Wechsel- bzw. Schwellspannungen sollen schließlich zu einer von den Fremdkörperkanten ausgehenden Rißbildung führen. Obwohl diese Hypothese vielfach zur Erklärung der Ausfälle von Wälzlagern herangezogen wurde, sind nur wenige Untersuchungsergebnisse veröffentlicht worden.

STYRI[66] vertritt die Ansicht, daß z.B. Schlackeneinschlüsse nur dann als Ausfallursache wirksam werden, wenn sie in der Tiefe der maximalen Beanspruchung liegen.

Umfangreiche Untersuchungen sind von BEAR UND BUTLER[3,11] durchgeführt wor-

den. Dabei hat es sich gezeigt, daß Fremdkörpereinschlüsse sowohl zu den Ermüdungsauffällen als auch zu der relativ großen Streuung solcher Versuche beitragen, wobei diese Einschlüsse die Rolle von spannungserhöhenden Stellen spielen, aber durchaus nicht alle aus derartigen Einschlüssen erzeugten Mikrorisse sich zu wirklichen Ermüdungsrissen ausweiten. Diese Mikrorisse verlaufen dabei meistens unter 45° zur Normalen; sie liegen damit in der Ebene der maximalen Schubspannung. Mit zunehmender Laufzeit werden in steigendem Maße Einschlüsse zur Rißbildung angeregt, so daß die Mikroriß-Dichte immer größer wird und die Möglichkeit, daß einzelne Mikrorisse zusammenlaufen und sich zu Ermüdungsrissen erweitern, wächst. Weiterhin wurde aus den genannten Untersuchungen deutlich, daß neben der Größe der Einschlüsse vor allem ihre Härte entscheidend ist für das Ausmaß des Auftretens von Mikrorissen. Dabei sind die weicheren Sulfide in geringerem Maße Ursache von Ermüdungsrissen als die härteren Oxyde und Karbide; am gefährlichsten sollen nach diesen Angaben Lösungen von Titannitriden und Titankarbiden sein. Im übrigen hat sich noch gezeigt, daß die Rißanfälligkeit wesentlich geringer wird, wenn die Versuche bei erhöhten Temperaturen durchgeführt werden.

Abb. 2(a). Faserverlauf in einer Kugel.

Bei der Herstellung der Rohlinge von Lagerringen treten infolge der verwendeten Verfahren oft ausgeprägte Faserrichtungen auf. Ähnlich ist es bei Kugeln und Rollen, die kalt geschlagen werden (s. Abb. 2). Diese Faserrichtung ist charakterisiert durch gleichmäßig angeordnete Zeilen von Einschlüssen aller Arten.

CARTER u.a.[6,8,11] haben den Einfluß dieser Faserrichtung ausführlich untersucht. Die Versuche wurden auf einem von MACKS[46] entwickelten Prüfstand durchgeführt, bei dem Kugeln durch ein Düsensystem angetrieben in einer zylindrischen Laufbahn laufen. Es wurden bei Kugeln die Lage der Grübchen zu der Faserrichtung durch entsprechendes Ätzen festgestellt.

Abb. 2(b). Faserverlauf in einem Ring.

Die Versuche haben ergeben, daß in den Polzonen, in denen die Fasern geschnitten werden, ein wesentlich größerer Anteil der Ermüdungsschäden liegt als bei einer gleichmäßigen Empfindlichkeit der ganzen Kugeloberfläche gegenüber Pittingbildung zu erwarten wäre. Gleichzeitig war die Lebensdauer der über die Pole gelaufenen Kugeln geringer als die der anderen. Neben diesen Untersuchungen an den Kugeln wurden auch die Laufbahnen von Ringen so bearbeitet, daß die Faserrichtung mit der Laufspur verschiedene Winkel einschloß. Hierbei zeigte sich wiederum, daß Pittings vorzugsweise an denjenigen Stellen auftraten, bei denen der Winkel zwischen Faserrichtung und Oberfläche für den betrachteten Ring am größten war. Ebenso war zu erkennen, daß die Lebensdauer derjenigen Ringe, deren Faserrichtung senkrecht zur Oberfläche liegt, kleiner ist als die der anderen.

Untersuchungen von SCHEIN[63] zeigten den gleichen Einfluß, wobei weiter festgestellt wurde, daß bei senkrechter Faserrichtung gegenüber der Lauffläche bei kleinerer Lebensdauer auch eine geringere Streuung der Versuchsergebnisse auftrat als bei parallel zur Oberfläche verlaufenden Fasern.

Die angeführten Veröffentlichungen zeigen, daß den Einschlüssen im Werkstoff eine gewisse Bedeutung beigemessen werden muß. Wenn sie auch bei weitem nicht ausschließliche Ursache von Ermüdungsschäden sind, so können sie doch je nach

ihrer Art, Größe und Lage Ausgangspunkte von Rissen sein, die schließlich zu Ausfällen führen. Eigene Erfahrungen bestätigen diese Ansicht nicht. Es wurden umfangreiche Untersuchungen[76] an gelaufenen Wälzlagern durchgeführt. Dabei wurde

Abb. 3. Von Einschlüssen unabhängige Rißbildung (a) geätzt (b) ungeätzt.

jeweils das Schadensgebiet mit Hilfe von Schliffbildern in der Nähe von Rissen nach Fremdkörpereinschlüssen abgesucht. Es konnten jedoch keine Anzeichen für einen ursächlichen Zusammenhang gefunden werden. Oft dagegen wurden in der Nähe von Rissen Einschlüsse festgestellt, die in keiner Weise die Rißbildung beeinflußten (s. Abb. 3). Auch aus direkt unter der Oberfläche liegenden Fremdkörpern (s. Abb.4) bildeten sich keine Risse oder Grübchen. Dieser Gegensatz zu den erwähnten Veröffentlichungen kann dadurch erklärt werden, daß viele Anzeichen dafür sprechen, daß sich Einschlüsse unter einer gewissen Größe nicht mehr nachteilig auf die Lebensdauer auswirken, und daß diese Grenze mit dem heutigen Wälzlagerstahl erreicht wird.

Abb. 4. Einschluß direkt unter der Oberfläche, der trotz hoher Beanspruchung nicht zum Ausfall führte.

Als weitere mögliche Einflußgröße wurde verschiedentlich die Oberflächengüte der Laufbahnen betrachtet. Hierbei geht man im allgemeinen von der Vorstellung aus, daß durch die Rauhigkeit der aufeinander abrollenden Flächen Spannungserhöhungen auftreten, die das Entstehen von Rissen begünstigen. Die hierzu bekannten Ergebnisse beziehen sich allerdings nicht auf vollharte (800 HB) Stähle, sie sollen jedoch vollständigkeitshalber erwähnt werden.

PETRUSSEWITSCH UND MISCHARIN[61,73] untersuchten den Einfluß der Oberflächengüte auf die Überrollfestigkeit. Die Versuche wurden angestellt mit gefrästen, gedrehten, geschliffenen, polierten und geläppten Rollen aus 0.45%igem Kohlenstoff-

stahl mit einer Härte von 300 HB. Es ergab sich eine wesentliche Verminderung in der Lebensdauer nur bei den gefrästen Rollen, während alle anderen Rollen ungefähr die gleiche Lebensdauer erreichten.

HELBIG[29,30] führte Versuche mit St. 42-Prüflingen verschiedener Oberflächengüte durch. Hierbei wurden Lebensdauerwerte gefunden, die bei den feingeschliffenen Proben um 11%, bei geläppten oder polierten Proben um 22% über der Lebensdauer der gedrehten Proben liegen, wobei zwischen polierten und geläppten Proben praktisch kein Unterschied in der Rollfestigkeit zu finden war.

Auch WAY[72] konnte feststellen, daß rauhere Oberflächen eher zu einer Pittingbildung neigen, gibt als Erklärung dafür allerdings an, daß in gröberen Oberflächen von der Bearbeitung her mehr Risse verhanden sind, die sich zu Grübchen erweitern.

Andere Untersuchungen[75] deuten auf den Einfluß von Restspannungen hin, die aus den verschiedenen Bearbeitungsvorgängen entstehen. Diese Restspannungen können sowohl Zug- als auch Druckspannungen sein und eine ziemliche Größe erreichen, verglichen mit den durch die äußere Belastung hervorgerufenen Spannungen. Diese Restspannungen können entweder allein — also bevor noch eine Belastung eintritt — oder im Verein mit einer an und für sich ungefährlichen Belastung zu Rissen führen, die sich dann, wie auch MATTSON[48] zeigt, erweitern und einen Ausfall herbeiführen, wobei die Änderung der Spannung pro Längeneinheit, d.h. der Spannungsgradient, eine entscheidende Rolle spielen soll.

SPANNUNGSVERTEILUNG UNTER DEM EINFLUSS ÄUSSERER BELASTUNGEN

Im Jahre 1895 ermittelte HERTZ[34] den elastischen Spannungszustand, der sich bei der Berührung zweier Körper unter Einwirkung einer äußeren Belastung in den Körpern einstellt. Damit war die Möglichkeit gegeben, die Beanspruchung des Werkstoffes bei Wälzlagern zu berechnen. Auf Grund dieser Tatsache wurden die verschiedensten Hypothesen über die Pittingbildung aufgestellt, ohne jedoch dem eigentlichen Problem der Ermüdung wesentlich näher zu kommen. Man suchte nach besonders kritischen Spannungen, die ausschlaggebend für Grübchenbildung sein sollten. So wurden die maximalen Druckspannungen in der Oberfläche, die Zugspannungen am Rande der Druckfläche oder im Inneren des betrachteten Körpers, die Schubspannungen unter der Oberfläche usw. als Ursachen für einen Ausfall des Lagers angesehen. Es ist klar, daß letzten Endes die äußere Belastung und der durch sie hervorgerufene Spannungszustand die Ermüdung hervorruft. Bevor man jedoch nicht den Vorgang der Ermüdung kennt, kann man sich auch nicht auf eine bestimmte Anstrengungshypothese festlegen.

LUNDBERG UND PALMGREN[45] nahmen die unter 45° gegen die Normale auf die Oberfläche gerichtete Hauptschubspannung im Inneren des Körpers zur Grundlage ihrer Lebensdauertheorie. Diese Annahme wird damit begründet, daß die im Inneren sich bildenden Risse unter 45° zur Oberfläche geneigt sind, ebenso wie die zugrundeliegende Hauptschubspannung.

PETRUSSEWITSCH[61] dagegen hält die zur Oberfläche parallele Schubspannung für gefährlicher, weist aber darauf hin, daß im Inneren des Werkstoffes weitaus höhere

Beanspruchungen ausgehalten werden als an der Oberfläche. Diese Ansicht wird damit begründet, daß unter der Oberfläche eine „Selbstheilung" von Mikrorissen eintreten kann, da hier aktive Stoffe von außen her keinen Zutritt haben.

WAY[72] erklärt das Aushalten von hohen Tiefenschubspannungen mit dem Vorhandensein allseitiger Druckspannungen, die ein Abgleiten erschweren.

KENNEDY[40] kommt auf Grund von Versuchsreihen mit geglühten und gehärteten Prüfkörpern zu dem Schluß, daß die maximale Schubspannung unterhalb der Oberfläche keineswegs die entscheidende Rolle bei dem Ermüdungsvorgang spielt, sondern höchstens in zweiter Linie die Pittingbildung beeinflußt. Er hält die Oberflächenbedingungen für wesentlicher.

KARAS[37] hat rechnerisch den Ort der größten Beanspruchung des Werkstoffes nach verschiedenen Anstrengungshypothesen untersucht. Dabei hat sich gezeigt, daß die Maxima nach den einzelnen Hypothesen etwa in der gleichen Tiefe unter der Oberfläche liegen, so daß auf Grund der Ermüdungserscheinungen nicht auf die Gültigkeit einer bestimmten Hypothese geschlossen werden kann. KARAS hält es für wahrscheinlich, daß in Wirklichkeit für die Anstrengung des Werkstoffes nicht der Spannungszustand in einem einzelnen, ausgezeichneten Punkt des Körpers maßgebend sein kann, sondern der innerhalb eines gewissen Gebietes. Diese Ansicht wurde auch schon von L. FÖPPL[19] vertreten, der das Integral der Gestaltänderungsarbeit längs einer Fließlinie als charakteristische Größe betrachtet.

Neben den Schubspannungen unter der Oberfläche und anderen Vergleichsspannungen werden oft auch aus der Reibung resultierende Oberflächenspannungen als entscheidend für eine Pittingbildung angesehen. Verschiedentlich sind dazu Untersuchungen angestellt worden. So fand NISHIHARA[55] bei Versuchen, bei denen ein gewisser Schlupf zwischen zwei aufeinander abrollenden Körpern erzwungen wurde, daß bei etwa 22% Schlupf die Lebensdauer am geringsten ist. Dafür wurde jedoch keine Erklärung gegeben. LOCATI UND FERRO[43] zeigten dagegen eine ständige Verminderung der Laufzeiten mit steigendem Gleitanteil. Von mehreren Seiten[2,36,55] wurden Haarrisse in der überrollten Oberfläche gefunden, die auf Oberflächenschubspannungen aus den Abrollverhältnissen hindeuten. Diese Risse waren dabei pfeilförmig ausgebildet und zeigten mit der Spitze in Gleitrichtung.

Wenn man die einzelnen Ergebnisse der verschiedenen Verfasser zusammenfassend betrachtet, so muß man sagen, daß bis jetzt noch kein schlüssiger Beweis vorliegt, daß eine bestimmte Beanspruchungshypothese dem Ermüdungsvorgang in jeder Hinsicht Rechnung trägt. Nach Art und Lage von Ermüdungsrissen, wie sie Abb. 5 in einer typischen Form im Schliffbild zeigt, kann nicht eindeutig zwischen den am meisten diskutierten Hypothesen—Schubspannung parallel zur Oberfläche oder Hauptschubspannung unter 45° zur Oberfläche—entschieden werden.

In diesem Zusammenhang sollen noch zwei Fragestellungen behandelt werden, und zwar die nach einer Ermüdungsfestigkeit und anderen Festigkeitswerten.

KARAS[38] und NISHIHARA[55] setzen eine Ermüdungsdauerfestigkeit proportional der Summe von Fließ- und Bruchspannung an, wobei aber bei beiden verschiedene Proportionalitätsfaktoren verwendet werden. Daneben wird nach MAILÄNDER (unter[38])

eine Proportionalität zwischen Dauerermüdungsfestigkeit und Brinellhärte angesetzt. Dem tritt MOORE (unter [38]) mit der begründeten Aussage entgegen, daß bei einer Härte von über 400 HB keine Grenzwerte der Ermüdungsfestigkeit mehr erwartet werden dürfen, da bei derartigen Stählen die plastische Verformbarkeit sehr gering

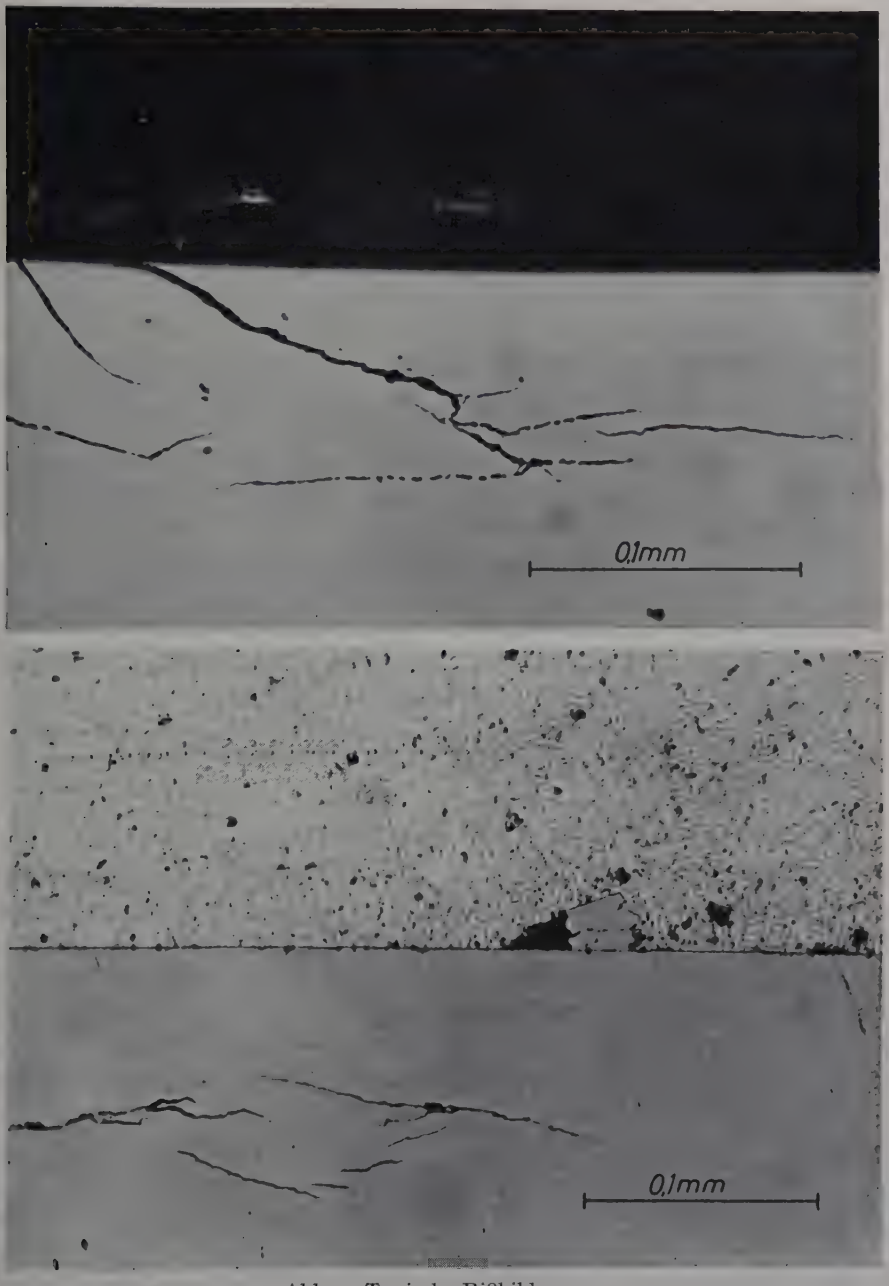


Abb. 5. Typische Rißbildungen.

ist. Auch TALBOURDET[67] konnte bei Überrollungsversuchen bis zu 500 Millionen Lastwechseln keine Dauerfestigkeit bei harten Stählen finden. Zu ähnlichen Ergebnissen kommt auch PETRUSSEWITSCH[61]. Die praktischen Erfahrungen der Wälzlagerindustrie[17,76] bestätigen diese Ansicht; im normalen Anwendungsbereich sind keine Anzeichen für eine Dauerfestigkeit festgestellt worden. Während JOHANSSON (unter[73]) gefunden hat, daß bei verschiedenen Werkstoffen kein Zusammenhang zwischen der zu einer bestimmten Lebensdauer gehörigen Spannung und der Bruchlast bzw. der Fließgrenze besteht, geht GOHN[26] noch weiter und behauptet, daß überhaupt keine genaue Wechselbeziehung zwischen der Ermüdungsfestigkeit und irgendeiner anderen, leichter zu bestimmenden Werkstoffeigenschaft existiert.

Die einzigen Vergleiche zwischen verschiedenen Festigkeitswerten an demselben Werkstoff sind von STYRI[66] veröffentlicht geworden. Er führte neben Wälzlagerversuchen Überrollungs-, Umlaufbiege- und Wechseltorsionsversuche durch. Dabei wurde der amerikanische Wälzlagerstahl 52 100 verwendet, der auf Härtewerte von 64 bis 17 HRc angelassen wurde. Aus den Ergebnissen zeigte sich, daß die Lebensdauer bei Kugellagern umgekehrt proportional ist der 9. Potenz der Spannung, bei überrollten Ringen der 12. Potenz, bei Umlaufbiegeversuchen der 15. Potenz und bei Wechseltorsionsversuchen der 10. Potenz der Spannung. Aus der ungefähren Gleichheit der Lebensdauerexponenten bei Kugellager- und Wechseltorsionsversuchen schließt STYRI, daß die Schubspannungen wahrscheinlich die Ursache der Ermüdungsschäden bei Wälzlagern sind.

Aus diesen unterschiedlichen Auffassungen über die Existenz einer Ermüdungsdauerfestigkeit und auch über einen Zusammenhang mit anderen Festigkeitswerten zeigt sich, daß man heute noch weit davon entfernt ist, hier etwas Endgültiges zu sagen. Eine rein empirische Betrachtung der Ausfallerscheinungen und Ausfallzeiten wird auch kaum zu einem Erfolg führen. Erst wenn man die eigentlichen Vorgänge, die zu einer Ermüdung führen, kennt, können vielleicht Parallelen gezogen werden. Hiervon wird noch weiter unten die Rede sein.

EINWIRKUNG DES SCHMIERMITTELS

Ein weiterer Faktor, der bei der Frage nach den Ursachen der Ermüdungserscheinungen betrachtet werden muß, ist das Schmiermittel. Neben den ersten Untersuchungen von WAY sind hierzu erst in letzter Zeit umfangreiche Versuche durchgeführt worden. WAY[72] ließ geschliffene Stahlrollen aufeinander abrollen und fand dabei, daß ohne Schmierung keine Grübchenbildung auftritt. Es bildet sich vielmehr eine Oxydhaut, die beim weiteren Überrollen wieder abgerieben wird. Allerdings soll nach WAY auch bei ungeschmiertem Lauf irgendeine Art latenter Zerstörung des Werkstoffes vorliegen, ohne daß darüber Näheres gesagt wird.

WAY ließ nämlich Rollen, die bereits ohne ein Schmiermittel gelaufen waren und keine Pittings zeigten, anschließend mit einem Schmiermittel laufen. Der Ausfall trat schon nach kurzer Zeit ein. Andererseits erweiterten sich Haarrisse, die an geschmierten Rollen nach längerem Lauf gefunden wurden, bei anschließendem trockenen Lauf nicht zu Pittings.

WAY sieht in den feinen pfeilförmigen Haarrissen am Ende der Druckfläche die

ersten Ermüdungserscheinungen. Aus Versuchen mit verschieden zähen Schmiermitteln kam er zu dem Schluß, daß es für jede Belastung wahrscheinlich eine bestimmte Grenze der Viskosität gibt, bei der keine Pittingbildung mehr erfolgt.

WAY erklärt nämlich die Wirkung des Öls auf die Grübchenbildung mit einem Eindringen des Öls unter Druck in feine Haarrisse und führt dazu folgende Punkte an:
Öl ist notwendig zur Pittingbildung; es muß eine hinreichend geringe Viskosität haben, um in die Haarrisse eindringen zu können; diese Risse erweitern sich unter dem Öldruck zu Grübchen, wenn sie eine dem Ölstrom entsprechende Richtung haben, die ein Eindringen des Öls erleichtert; rauhere Oberflächen führen eher zur Grübchenbildung, da hierbei mehr Oberflächenrisse von Natur aus vorhanden sind. Im Gegensatz zu WAY behauptet NISHIHARA[55], daß die Ermüdungsfestigkeit mit kleiner werdender Viskosität ansteige.

In letzter Zeit sind weitere, grundlegende Versuche von BARWELL UND SCOTT[2] durchgeführt worden. Hierbei wurden verschiedenartigste Schmiermittel mit Hilfe eines modifizierten Vierkugelapparates hinsichtlich ihres Einflusses auf die Lebensdauer untersucht. Dabei ergab sich folgendes:

Die Pittingbildung begann mit feinen, quer zur Rollrichtung verlaufenden Rissen an der Oberfläche. Bei längeren Laufzeiten traten Zusammenballungen von derartigen Rissen auf, die dann schließlich zu einer Pittingbildung führten. Die Vergrößerung der Pittings erfolgte bei verschiedenen Schmiermitteln unterschiedlich. Bei Mineralölen trat eine plastische Verformung der Ränder und ein Glätten der Kanten auf; bei nichtbrennbaren Hydraulikflüssigkeiten vergrößerten sich die Grübchen sehr rasch, und es kam manchmal zu Schalenbrüchen. Bei einigen Schmiermitteln ergab sich dabei die 3. Potenz für die Abhängigkeit der Lebensdauer von der Belastung, so daß zwar die Lebensdauer, aber nicht das Potenzgesetz zwischen Lebensdauer und Belastung beeinflußt wird. Bei Mineralölen und Flüssigkeiten auf Polyalkalen–Glykol-Basis ließ sich ein Steigen der Lebensdauer mit der Viskosität finden, bei anderen Schmiermitteln jedoch nicht. Vergleichbar waren in jedem Fall nur gleichartige Schmiermittel. Eine wesentliche Verminderung der Lebensdauer ergaben nichtbrennbare Hydraulikflüssigkeiten auf Wasser–Glykol-Basis. Ähnliche Erfahrungen wurden auch an Pumpenlagern gemacht. BARWELL UND SCOTT hielten es für wahrscheinlich, daß weniger die Viskosität des Schmiermittels als vielmehr dessen Kompressibilität und die Änderung der Viskosität mit dem Druck die Ermüdungsfestigkeit beeinflussen.

CORDIANO u.a.[13] führten Versuche mit Schrägkugellagern durch. Auch hier zeigte sich eine wesentliche Beeinflussung der Lebensdauer durch die verwendeten Schmiermittel. Hydraulikflüssigkeiten auf Wasser–Glykol-Basis verminderten die Lebensdauer wesentlich.

PETRUSSEWITSCH[61] berechnete die Spannungsverteilung bei reinem Rollen unter Berücksichtigung eines hydrodynamischen Schmierfilms und erhielt nur geringe Abweichungen gegenüber HERTZ, so daß hiermit die Grübchenbildung kaum begründet werden kann. Die von ihm angeführten Versuche von TRUBIN zeigten, daß bei einer Verringerung der Viskosität, erreicht durch eine Temperaturerhöhung von 45° auf 82°C, ein Abfall der Ermüdungsfestigkeit um 22% erfolgte.

Versuche von OTTERBEIN[58] mit Rollenlagern zeigten ebenfalls eine Lebensdauerminderung bei Abnahme der Viskosität; gleichzeitig wird erwähnt, daß aber noch andere Eigenschaften des Schmiermittels eine Rolle spielen. Die bei dem gleichen Autor angeführten Versuche von BARNES UND RYDER ergeben eine Proportionalität zwischen Viskosität und Lebensdauer.

Weitere umfangreiche Versuche wurden von CARTER[7,9,10] durchgeführt. Bei vier

handelsüblichen Paraffinölen verschiedener Viskosität ergab sich zwischen Lebensdauer und Viskosität ein Potenzgesetz mit dem Exponenten 0.2. Die Streuung blieb dabei ungefähr gleich.

Eine zweite Versuchsreihe mit chemisch verschiedenen Schmiermitteln gleicher Viskosität ergab keine Anhaltspunkte für eine Beziehung zwischen der Lebensdauer und irgendwelchen chemischen Eigenschaften, sondern zeigte die maßgebende Rolle der physikalischen Eigenschaften. So ergab sich, daß die Abhängigkeit der Viskosität vom Druck einen entscheidenden Einfluß auf die Lebensdauer hat. CARTER vermutet weiterhin, daß das Verhalten des Schmiermittels bei plötzlichen Druckänderungen, wie sie beim Überrollvorgang vorkommen, einen Einfluß hat. Er nimmt dabei an, daß sich die Viskosität nicht spontan dem Druck anpaßt, sondern erst mit einer gewissen Verzögerung.

Zusammenfassend muß also trotz der sich widersprechenden Einzelergebnisse ein wesentlicher Einfluß des Schmiermittels auf die Lebensdauer angenommen werden.

METALLURGISCHE ERSCHEINUNGEN

Umfangreiche Versuche über die metallurgischen Vorgänge während der Laufzeit von Wälzlagern wurden von JONES[36] durchgeführt. Er fand bei Wälzlagerringen, die mehrere hundert Stunden unter Flächenpressungen von 300 bis 500 kg/mm² gelaufen

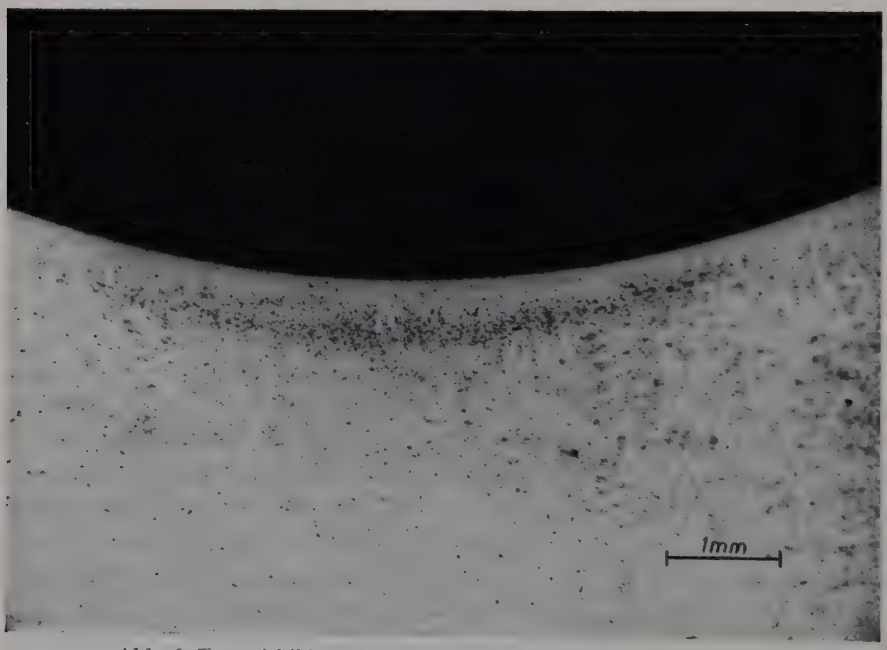


Abb. 6. Troostitbildung in der am höchsten beanspruchten Zone.

waren, durch Ätzen von Schliffen senkrecht zur Laufbahn Gefügeumwandlungen unter der Oberfläche (s. Abb. 6). Aus dem ursprünglich feinkörnigen Martensit hatte sich teilweise Troostit gebildet. Mit wachsender Belastung und steigender Last-

wechselzahl vermehrte sich die Troostitausscheidung, die auf das Gebiet maximaler Anstrengung unter der Laufbahnoberfläche begrenzt war. Wenn nach längerer Laufzeit die Troostitbildung einen Höhepunkt erreicht hat, treten in der Ebene der großen Druckellipsenachse parallel zur Oberfläche verlaufende Linien auf, die in der Ebene der kleinen Druckellipsenachse schräg zur Oberfläche verlaufen.

JONES erklärt diese Vorgänge damit, daß die Schubspannungsenergie teilweise in innere Reibung überführt wird und so örtliche Temperaturspitzen zur Folge hat, die dann eine Umwandlung des Martensit in Troostit bewirken. Die beobachteten Linien werden gedeutet als Gebiete plastischen Fließens, hervorgerufen durch die Hauptschubspannungen, die größer sind als die durch die Troostitbildung herabgesetzte Fließgrenze. Wenn einmal ein örtliches Fließen eingesetzt hat, kommt es bald zu größeren Gleitungen, die die Zerstörung des Werkstoffes hervorrufen. Bei diesen Versuchen wurde weiterhin festgestellt, daß bei Ringen, die im Anfangszustand schon geringe Mengen Troostit enthielten, die auf die Überrollung zurückzuführende Troostitbildung wesentlich schneller verläuft und die Lebensdauer mindert.

BARWELL UND SCOTT[2] fanden bei ihren Untersuchungen ähnliche Gefügeumwandlungen, sagen aber dazu, daß diese erst bei derart hohen Belastungen auftreten, die in der Wälzlagertechnik außerhalb des normalen Anwendungsbereiches liegen.

PETRUSSEWITSCH[61] meint dazu, daß im normalen Belastungsbereich die Umwandlungsgebiete so klein sein können, daß sie visuell nicht mehr feststellbar sind und ein Härteabfall nicht meßbar ist.

BEAR UND BUTLER[3] fanden bei Überrollversuchen Strukturänderungen in der am meisten beanspruchten Zone nur bei erhöhten Temperaturen (etwa 120°C). Zwischen den Troostit-Zonen und der überrollten Oberfläche war ein Streifen, in dem keinerlei Gefügeänderung auftrat. Nach längeren Laufzeiten traten in der Troostit-Zone feine Risse auf, die etwa 45° gegen die Oberfläche geneigt waren. Diese Risse vergrößerten sich mit der Zeit. Das nach längerer Laufzeit auftretende Pitting reicht meistens nicht in die Troostit-Zone hinein. Diese Tatsache wird dahingehend gedeutet, daß die härtere Oberflächenschicht wegen der ungenügenden Abstützung durch die weichere Troostit-Zone bricht. Dazu kommt, daß in manchen Fällen bei gelaufenen Wälzlagern[75] eine verfestigte Oberflächenschicht gefunden wird, die diesen Effekt noch verstärken könnte.

Auf Grund metallurgischer Untersuchungen ist in letzter Zeit versucht worden, die Ermüdungserscheinungen mit Hilfe der Versetzungstheorie zu beschreiben und eventuell rechnerisch zu erfassen. Aus der großen Anzahl der über dieses Thema veröffentlichten Untersuchungen sollen nur einige herausgegriffen werden, um zu zeigen, in welche Richtung die einzelnen Vorstellungen gehen.

Wenn Versetzungen auf ein Hindernis stoßen, kann der dadurch bewirkte Stau zur Bildung eines Risses senkrecht zur Gleitebene der Versetzungen führen, wie OROWAN[26] gezeigt hat. Weiterhin ist untersucht worden, wie sich eine Versetzungsanordnung in einem Spannungsfeld verhält, wobei festgestellt wurde, daß sich der Riß dann in Richtung der Gleitebene öffnen kann. Auch von STROH (unter[68]) sind ähnliche Berechnungen durchgeführt worden. Dabei ergab sich, daß sich die Energie einer bestimmten Versetzungsanordnung durch die Bildung eines Risses vermindern kann.

THOMPSON[68] sagt dazu jedoch, daß dieser Mechanismus wahrscheinlich nicht für Ermüdungsrisse anwendbar ist, weil hierbei stärkere plastische Verformungen fehlen und auch die Risse nicht in Richtung der Gleitebene liegen, wie in der Praxis festgestellt werden konnte. In diesem Zusammenhang verweist THOMPSON auf die Hypothese von MOTT für das flächenzentrierte, kubische Gitter: Wenn sich Schraubenversetzungen längs ihrer Gleitebene durch ein Kristallgebiet mit anderen Schraubenversetzungen bewegen, entstehen beim Kreuzen der Versetzungen Gitterfehlstellen. Wenn nun durch eine Umkehr der Belastungsrichtung die Versetzungen gezwungen werden, sich in der entgegengesetzten Richtung zu bewegen, werden sich die schon vorhandenen Fehlstellen nicht wieder auffüllen; es werden vielmehr an anderer Stelle neue Fehlstellen entstehen. Somit ist bei alternierender Beanspruchung mit einer fortlaufenden Erzeugung von Gitterfehlstellen zu rechnen, wodurch das Gitter allmählich zerstört wird. Wenn diese Zerrüttung ein gewisses Maß erreicht hat, wird eine Rekristallisation einsetzen. Dabei findet eine Volumenkontraktion statt, und die dabei entstehenden Zugspannungen weiten die durch Aneinanderlagerung von Gitterfehlstellen entstandenen Mikrorisse auf. Tritt dieser Vorgang an der Oberfläche auf, dann kann durch das Eindringen von Gasen die Rißbildung verstärkt werden. Da im Anfangszustand diese Vorgänge auf den Bereich einzelner Körner begrenzt sind, ist die Korngröße für die Lebensdauer wesentlich.

Wenn man die vorliegenden Ergebnisse über die metallurgischen Erscheinungen zusammenfassend betrachtet, dann kann man sagen, daß auf diese Weise wohl am ehesten ein Verständnis des Ermüdungsvorganges und der entsprechenden Ursachen erreicht werden kann. Wie weit dabei die Versetzungstheorie in ihrer jetzigen Form herangezogen werden kann, bleibt abzuwarten.

SCHLUSSFOLGERUNGEN

Betrachtet man die zur Frage der Ermüdungserscheinungen bei Wälzlagern vorliegenden Ergebnisse, dann kann man feststellen, daß sich trotz des relativ umfangreichen Materials noch keine einheitliche Auffassung über das Auftreten von Pittings und die einzelnen Einflüsse gebildet hat. Wenn man als Ziel aller dieser Untersuchungen eine umfassende Theorie der Ermüdung von Wälzlagern ansieht, mit Hilfe derer die Lebensdauer berechnet werden kann, dann sind hierzu zwei Wege möglich: Der eine geht aus von empirischen Daten, die in geeigneter Weise verallgemeinert werden; der andere baut auf Erkenntnissen über den Ermüdungsvorgang auf und versucht auf diese Weise zu praktisch verwertbaren rechnerischen Beziehungen zu gelangen. Es ist ohne weiteres klar, daß der erste Weg der einfachere ist und schneller zu praktisch verwertbaren Aussagen führt. Deshalb sind auch die meisten Untersuchungen in diese Richtung gegangen.

Als wesentlicher Punkt ist dabei die Abhängigkeit der Lebensdauer von der äußeren Belastung untersucht worden. Der Ausgangspunkt dafür waren Lebensdauerversuche mit Wälzlagern, die mit verschiedenen Belastungen durchgeführt wurden.

Einerseits wurde damit bei konstant gehaltenen Berührungsverhältnissen die Lastabhängigkeit der Lebensdauer bestimmt, andererseits wurde versucht, Aussagen über die Abhängigkeit

der Lebensdauer von den Berührungsverhältnissen zu erhalten, die bei konstanter äußerer Belastung für die Spannungsverteilung in den sich berührenden Körpern maßgebend sind. Wie oben gezeigt, ging man dabei von sehr verschiedenen Beanspruchungshypothesen aus, die aber alle auf den durch die HERTZ'sche Theorie berechenbaren Spannungen basieren. Zu diesem Vorgehen ist zweierlei zu sagen: Einmal ist die Wahl einer bestimmten Beanspruchungshypothese mehr oder weniger willkürlich, zum anderen sind die Voraussetzungen, die HERTZ seiner Theorie zugrunde gelegt hat, sicherlich nicht bei allen in Wälzlagern vorkommenden Berührungsverhältnissen gegeben. Dadurch wird bei Vergleichsuntersuchungen immer ein Unsicherheitsfaktor auftreten.

Von den verschiedenen Anstrengungshypothesen scheinen in letzter Zeit nur noch zwei hauptsächlichst diskutiert zu werden, nämlich die *Hauptschubspannungshypothese* und die *Schubspannungsenergiehypothese*. Welche von beiden den wirklichen Verhältnissen näher kommt, wird nur schwer zu entscheiden sein.

Nächst der Belastung dürfte das *Schmiermittel* den größten Einfluß auf die Lebensdauer der Wälzlager haben. Trotz mancherlei Widersprüchen zeigen die vorliegenden Versuchsergebnisse doch deutlich, daß das Schmiermittel einen wesentlichen Einfluß auf die Lebensdauer hat. Dieser Einfluß wird umso mehr zu beachten sein, je mehr man auf die Verwendung synthetischer Öle übergeht, über die noch zu wenig Erfahrungen vorliegen.

Nach den Untersuchungen scheint festzustehen, daß zumindest in erster Näherung der Schmiermitteleinfluß unabhängig ist von der Belastung, und daß bei gleichartigen Ölen die Viskosität die ausschlaggebende Rolle spielt. Damit läßt sich aber noch nichts über verschiedenartige Schmiermittel aussagen. Wie weit die Druckabhängigkeit der Viskosität für das Verhalten des Schmiermittels mit herangezogen werden muß, kann zur Zeit noch nicht endgültig gesagt werden. Dazu sind größere, systematisch angelegte Versuchsreihen erforderlich. Sicher ist der Einfluß des Schmiermittels auf die Lebensdauer so groß, daß er nicht vernachlässigt werden darf, zumal andere Einflüsse äußerst genau in die Berechnung einbezogen werden.

Eine der großen Schwierigkeiten bei der Auswertung von Lebensdauerversuchen mit Wälzlagern besteht darin, daß die Ergebnisse stark streuen, was eine große Anzahl von Versuchen erforderlich macht, um mit *statistischen Verfahren* sichere Aussagen machen zu können. Von verschiedenen Seiten[5,7,9,10,22,23,27,44,45] wird für den Ausfall von Wälzlagern die Weibull-Verteilung angesetzt, die sich in der Praxis gut bestätigt hat. Die mit allen statistischen Verfahren verbundene Unsicherheit hat allerdings zur Folge, daß bei Vergleichsuntersuchungen kleine Differenzen nicht immer richtig erkannt werden können.

Die Streuung der Ergebnisse wird im allgemeinen der Inhomogenität des Werkstoffes zugeschrieben; sie kann aber durch einfachere Versuchsbedingungen vermindert werden. So hat sich gezeigt[2,5,11,44,46], daß mit entsprechenden Prüfständen, auf denen nur *Wälzlagerteile* geprüft werden, geringer streuende Ergebnisse erhalten werden. Gleichzeitig werden die Versuchszeiten wesentlich kürzer, so daß sich viele Untersuchungen, die mit vollständigen Wälzlagern aus Zeitgründen nicht angestellt werden können, auf diese Weise durchführen lassen.

Auf Grund der Schwierigkeiten, die Lebensdauerversuche mit Wälzlagern mit sich bringen, hat man nach Zusammenhängen zwischen der Lebensdauer und leichter bestimmbaren *Wechselfestigkeiten* gesucht. Die hierzu vorliegenden Ergebnisse geben kein klares Bild. Das liegt zum großen Teil daran, daß Werte, die an ungehärteten

Prüfstücken gefunden wurden, auf gehärtete übertragen werden. Schon die Unterschiede im Bruchverhalten sind so groß, daß derartige Schlüsse fragwürdig sind.

Etwas günstiger dürften die Verhältnisse liegen, wenn man verschiedene Festigkeitswerte von Werkstücken gleicher Härte in Beziehung zu Lebensdauerversuchen mit Wälzlagern setzt. Hierbei bestehen sicherlich irgendwelche Zusammenhänge. Es bedarf jedoch erst genauer und umfangreicher Versuche, bis man auf empirischer Grundlage entsprechende Vergleiche ziehen kann; andernfalls gerät man leicht in die Gefahr, zu falschen Schlüssen zu kommen. Die Spannungsverteilung beim Wälzvorgang unterscheidet sich wesentlich von derjenigen bei allen anderen Festigkeitsversuchen. Die oft gezogene Parallele zu Wechseltorsionsversuchen entspringt offensichtlich der Ansicht, daß in beiden Fällen die Schubspannung für den Ausfall des Prüfstückes ausschlaggebend ist. Während aber beim Überrollvorgang neben den Schubspannungen sehr hohe Normalspannungen auftreten, ist dies bei reiner Torsion nicht der Fall. Außerdem spielen beim Überrollen sicher die aus dem Zusammenwirken von zwei Körpern resultierenden Oberflächenbedingungen eine Rolle, die beim Torsionsversuch auch entfallen.

Es wäre daher vorteilhaft, wenn ähnlich den anderen Wechselfestigkeiten eine *Wälzfestigkeit* entsprechend den in Wälzlagern vorkommenden Verhältnissen allgemein eingeführt werden würde, wie sie schon von NIEMANN[53] verwendet wird. Dabei muß darauf Rücksicht genommen werden, daß die Prüfkörper einfach gestaltet sind, die Versuchsvorrichtung keinen unkontrollierbaren Einfluß auf die Prüfkörper ausübt, die Versuchsbedingungen klar festgelegt und eingehalten werden können und die Versuchszeiten in einer vernünftigen Größenordnung liegen. Eine so ermittelte Wälzfestigkeit einer geometrisch, werkstoffmäßig und in ihren Betriebsbedingungen bestimmten Wälzpaarung könnte dann mit anderen Festigkeitswerten verglichen werden.

Während bisher die makroskopischen Verhältnisse bei der Ermüdung von Wälzlagern betrachtet wurden, müssen bei dem zweiten oben angedeuteten Weg die mikroskopischen Eigenschaften des Werkstoffes herangezogen werden. Man weiß heute, daß bei der Beanspruchung eines Werkstoffes von einem bestimmten Zeitpunkt an in den einzelnen Kristallen und Kristalliten *Mikrogleitungen* auftreten, die zumindest teilweise irreversibel sind und somit zu örtlichen plastischen Verformungen und Temperaturspitzen führen. Mit steigender Lastwechselzahl steigert sich die Bildung von Mikrogleitungen, bis schließlich ganze Gleitbänder sichtbar werden. Die durch die Zerrüttung des Werkstoffes in diesen Gleitbändern auftretenden Mikrorisse können sich unter dem Einfluß des vorhandenen Spannungsfeldes der äußeren Kräfte zu größeren Rissen ausweiten, die schließlich zu einem Ausfall des Prüfstückes führen.

Wenn in dem eben Gesagten nur allgemein von Beanspruchungen des Prüfstückes die Rede war, ohne genauer zu sagen, in welcher Weise diese Beanspruchung aufgebracht wurde und welcher Spannungszustand sich ausbildete, dann geschah das deshalb, weil nach den bis jetzt vorliegenden Erkenntnissen die Zerstörung des Werkstoffes im mikroskopischen Gebiet bei allen Beanspruchungsformen ähnlich verlaufen dürfte. An dieser Stelle wäre also die Möglichkeit gegeben, anstelle von Wälzversuchen auch andere Dauerversuche zu betrachten, um einen genaueren Einblick in die Vorgänge, die zur Bildung von Rissen führen, zu erhalten.

Die bis jetzt durchgeführten Untersuchungen auf diesem Gebiet beschränken sich meistens auf qualitative Auswertungen von Versuchen, ohne daß es bereits gelungen wäre, allgemein gültige quantitative Aussagen zu treffen. Zur rechnerischen Erfassung der geschilderten Vorgänge ist es notwendig, Modellvorstellungen zu entwickeln, mit Hilfe derer in Anlehnung an die Versuchsergebnisse eine Hypothese über das Ermüdungsverhalten aufgebaut werden könnte. Die Versetzungstheorie könnte hier ein wertvolles Hilfsmittel sein.

Zusammenfassend kann wohl behauptet werden, daß mit einer Erkenntnis des Ermüdungsvorganges noch viel erreicht werden kann auf dem Wege zu einem Berechnungsverfahren für die Lebensdauer von Wälzlagern. So lange jedoch diese Erkenntnisse noch nicht vorhanden sind, muß der empirische Weg weiter beschritten werden, wobei allerdings mehr als bisher alle wesentlichen Einflüsse systematisch untersucht werden sollten.

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A RAILWAY ACCIDENT

A CONDENSED VERSION OF THE BRITISH MINISTRY OF TRANSPORT & CIVIL AVIATION REPORT* ON THE DERAILMENT WHICH OCCURRED ON 12th AUGUST 1958 AT BOROUGH MARKET JUNCTION IN THE SOUTHERN REGION, BRITISH RAILWAYS

SUMMARY

Report on a derailment due to an excessively worn switch and stock rail.

DERAILMENT

The 6.52 a.m., 10-coach, Up electric train from Sanderstead to Cannon Street became derailed at the eighth coach as it travelled slowly over a set of badly worn switches on the Up local line where the routes to Charing Cross and Cannon Street diverge. The eighth coach was much shaken as a result of the derailment and was forced against the parapet of the viaduct, and the ninth coach was separated from it and slewed across the lines leading to Charing Cross, and tilted, but none of the few passengers in these coaches was seriously injured. Six were taken to hospital suffering from minor injuries but were not detained. The emergency call for help was answered very promptly and the fire services, ambulances, and police reached the site within a few minutes. The circuit breakers in the power supply lines to the conductor rails tripped properly, and the power on adjacent lines was cut off within a minute.

DESCRIPTION

Train frequency

Borough Market Junction is a very busy junction with heavy traffic on all lines. On the Up local line the number of trains each weekday over No. 26 points is about 210 towards Metropolitan Junction and Charing Cross, and about 65 towards Cannon Street. With this frequency permanent way renewals, except very minor ones, are not possible, and such work is carried out at weekends, when traffic is easier and "track occupations" can be given to the engineering staff.

Permanent way details and damage

Points No. 26 consisted of 95 lb. bull-head carbon-steel rails on wood crossing timbers. The "E" type curved switches and stock rails were of "joggled" design, in

* Colonel W. P. Reed, reporter. Published November 1958 (London, Her Majesty's Stationery Office), condensed by F. T. Barwell. Published in *Wear* by permission of the Controller of Her Britannic Majesty's Stationery Office.

which the toe of the switch is given a substantial thickness amounting to $\frac{3}{8}$ in. and the stock rail is kinked immediately in front of the toe to that extent so as to provide a smooth alignment on the running edge from stock rail to switch when the switch is closed. This design is frequently used at facing points where traffic is dense and where wheel flanges tend to bear heavily against the switch rails, since it allows much more metal at the thin end of the blade towards the tip. Manganese steel switches are authorised for these points because of the rapid wear which takes place on carbon steel switches, and replacement sets in manganese steel had been on order for some months. There had, however, been delay in the assembly of the curved switches and stock rails at the Southern Region permanent way depot due to the very great amount of work there in connection with the modernisation programme, and the previous manganese switches had been replaced with carbon steel switches and stock rails on 15th March 1958. The frog of the crossing, which, as will be apparent from the sketch, is part of a compound assembly and therefore of special design was of cast-manganese steel.

The points were not damaged by the derailment, and score marks were visible on the left hand switch rail to show where wheel flanges had travelled over it and had dropped outside near the heel, where the first broken chair was found. There were a number of broken or displaced chairs, sleepers and conductor rails in the lead of No. 26 points and on the two turnouts ahead on the Charing Cross lines, but the crossing assembly was fortunately undamaged.

The switch rail was excessively "side-cut", and the top of it for about the first 2 feet from the end was chipped, leaving a flattened top instead of one sloped to fit against the stock rail. The stock rail was also excessively side-cut and the joggle had completely disappeared on the running edge so that the alignment was regular along the length of this rail on the route leading towards Charing Cross which, as already indicated, carries over three times as much traffic as the other route. As a result, when the points were closed for the Cannon Street route, the toe of the left hand switch projected inside the running edge of the rail and the flattened top provided a sort of ramp for the flanges of oncoming wheels, which would have been bearing against the left hand rail because of the curvature, to ride up it on to the rail table.

The photograph (Fig. 1), which was taken after the switch and stock rail had been removed from the track and reassembled, shows the heavy side wear and the even alignment of the running edge on the worn stock rail with no trace of joggle, though a vestige can be seen at the bottom of the head just clear of the toe of the switch. The ramp effect at the end of the switch, caused by battering can also be seen in the detailed photograph.

REPORT

Motorman M. C. Chandler, who was in charge of the 6.52 a.m. electric train, said that on leaving London Bridge the aspect of the starting signal was a single yellow; the next signal, No. D.27 at Borough Market Junction, also showed a single yellow.

He had shut off power before reaching this signal and he coasted past it at about 15 m.p.h. as he expected to be stopped at the next signal. C.109 at Cannon Street, and his view of it would be short on account of the curvature. The signal came into view at the usual point, when he was about 120 yards from it, and as it was showing a green aspect he applied power. Subsequent measurements showed that at this moment the eighth coach would have been over points No. 26 with the trailing motored bogie a few yards from them.

Ganger A. A. Grover, who had been in charge of permanent way maintenance here for the past four years, said that he had last inspected points No. 26 on the Sunday, two days before the accident. He found a sliver about 2 in. long broken off the top of the switch rail and came to the conclusion that the generally worn state of the switch

Fig. 1. Photograph of the worn and damaged rail (Courtesy of British Railways).

and stock rail was such that it ought to be changed. He reported this to his permanent way inspector early on Monday morning and was told that arrangements would be made to change it during the next weekend.

Permanent Way Inspector E. K. Drane said that he also had looked at the points on the Sunday before the accident, though not in company with Ganger Grover, and had already decided that the switch must be changed when Grover spoke to him on the next day. He had been keeping a watch on the wear at these points for some little time because he knew that they must soon be renewed, but he was not unduly concerned about their condition when he saw them on the Sunday. He said that they were—

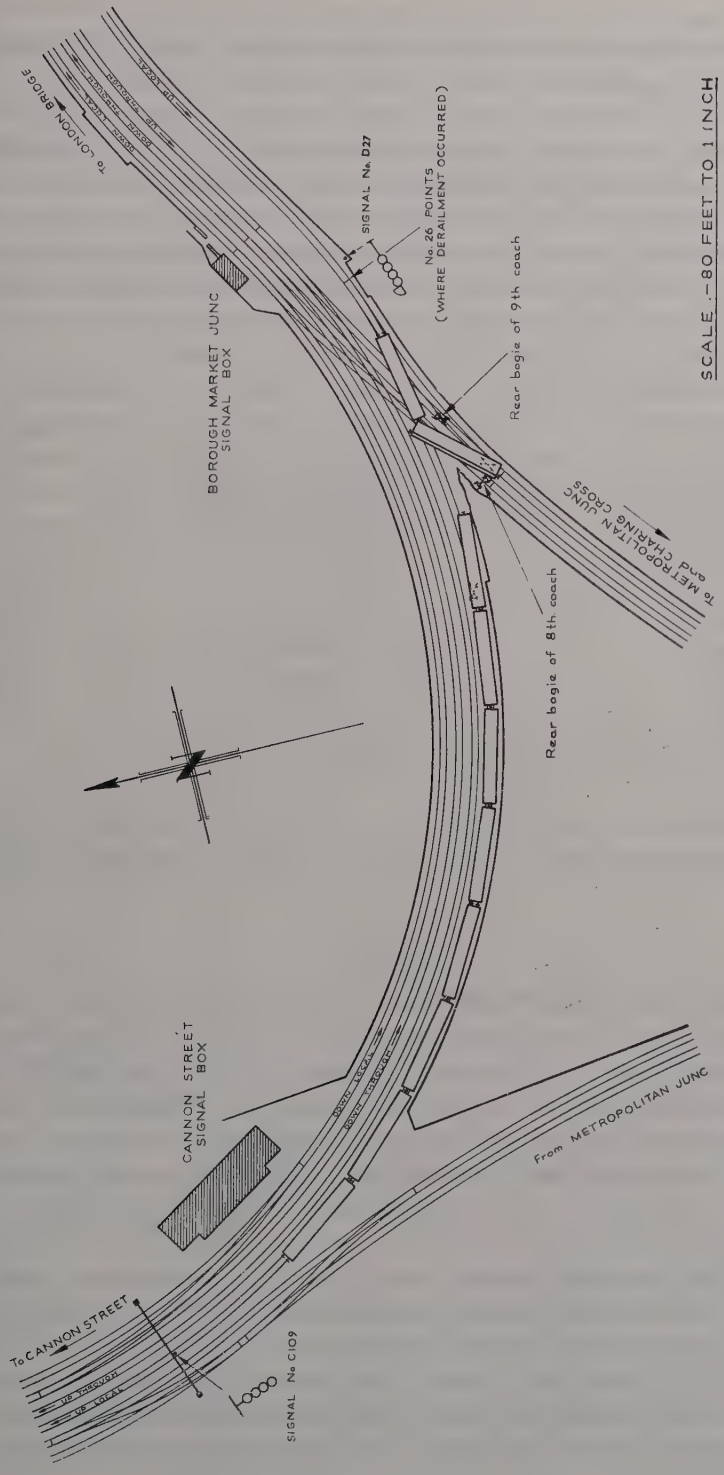

Fig. 2. Layout of the tracks and the position of the relevant signals.

"just the same as a lot more on the section, you go round and notice these jobs want doing or getting near to the time when they want to be done; that was more or less as I looked at that one."

Mr. Drane added that he always paid particular attention to these points because of the exceptionally heavy wear to which they were subjected; before the last change of switches on 15th March 1958 the previous set of manganese switches had been laid 9 months earlier, and the manganese set before that had only lasted 10 months.

In view of Inspector Drane's insistence on the attention which he gave to this turnout because of the rapid wear which occurred, I asked how often the points had been renewed in recent years, and was given the following information:—

Date renewed	*Type of Material*	*Life in months*
13th April 1947	Manganese Steel	22
6th February 1949	Manganese Steel	21
12th November 1950	Manganese Steel	22
7th September 1952	Manganese Steel	17
28th February 1954	Carbon Steel	13
20th March 1955	Manganese Steel	17
19th August 1956	Manganese Steel	10
23rd June 1957	Manganese Steel	9
16th March 1958	Carbon Steel	5
12th August 1958 (after the accident) ..	Carbon Steel	

It was also stated that since the introduction of 10-car electric trains in March 1957, followed by 12-car diesel trains in June 1958, the rate of wear had appreciably increased.

CONCLUSION

As I have already stated, the derailment, was due to an excessively worn switch and stock rail. The condition of these points had not been overlooked by Permanent Way Inspector Drane but he made a serious error of judgment in not renewing them some time earlier. It may be that he had allowed his judgment to be influenced by past experience of the rate of wear of permanent way here, and had been reluctant to renew the switch and stock rail earlier as they had been in the track for so short a time; this cannot, however, be regarded as an excuse for failing to deal with the situation on its merits.

There were two interesting features in the wear on the switch and stock rails, which are worthy of comment. The first was the side cutting of the stock rail throughout its length, which as already stated was on a right handed curve of 12½ chains radius, caused by the heavy traffic towards Charing Cross. This wear on the running edge meant that the top of the switch rail had no lateral support when closed against the stock rail for the Cannon Street route. The other feature was the complete dis-

appearance of the joggle on the running edge of the stock rail which, combined with the side cutting, exposed the blunt end of the switch rail and allowed it to be battered by the flange of the leading outer wheel on each bogie on every train travelling towards Cannon Street. The flanges of such wheels would all have been bearing hard against the outer rail of this sharp curve, and the sideways pressure against the unsupported head of the switch where it was thinnest towards the nose, combined with the end batter on the exposed nose by the wheel flanges, undoubtedly caused the head to be chipped and broken away over about the first 2 feet of the switch rail, so that a rough ramp was developed.

The wear must have reached a critical stage when the rear bogie of the eighth coach came to the points, and it may also be that the broken piece from the top of the switch rail which Inspector Drane found, became detached shortly before or as this bogie passed over it; the rough fractured surface left on the switch rail would have provided further grip for the riding wheel flange. The mounting of the flange over the ramped head on to the rail table may, in addition, have been assisted by the application of power which would have enhanced the tendency to climb.

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Wear, 3 (1960) 144–149

Brief Note

THE INFLUENCE OF WORKHARDENING ON THE COEFFICIENT OF FRICTION

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In 1950 PARKER AND HATCH[1], and MCFARLANE AND TABOR[2] demonstrated, independently, that if a soft metal is in contact with a hard surface, under a normal load, and a tangential force is subsequently applied, then the area of contact increases. MCFARLANE AND TABOR accounted for this increase by suggesting that the normal and tangential stresses were interacting and, consequently, that the criterion for plastic yielding of the soft metal was determined by the combined stresses. An analysis was given which utilised a yield criterion of the same form as that of Von Mises. Recently[3] it was shown that this analysis could apply only to an ideally plastic material or, what approximates to this in practice, a fully workhardened metal and that for such a material the coefficient of friction would be independent of the normal load. MCFARLANE AND TABOR[9], however, had found that the coefficient of friction of a steel ball sliding on indium in the load range 0–150 g was markedly dependent on the normal load, the coefficient of friction reaching very high values at low loads. This suggested that the behaviour of indium in this load range was not even approximately that of an ideal plastic material and this was subsequently verified when the micro-hardness of indium was determined using loads ranging from 0–100 g, under which conditions indium was shown to workharden. With the aid of a few simple assumptions, the yield criterion proposed by MCFARLANE AND TABOR[2] was amended to allow for workhardening and an analysis was given which, it was suggested, was applicable to a material which was incompletely workhardened when slip occurred. This analysis was applied to indium and was shown to account for that dependence of the coefficient of friction on the normal load which had been observed by MCFARLANE AND TABOR.

More recently, TABOR[4] has extended the MCFARLANE AND TABOR analysis to account for the influence of surface contamination on the coefficient of friction of metals but again, the analysis refers to ideally plastic materials and ignores the influence of workhardening. Consequently, it can apply only to cases where Amontons's laws are obeyed and it cannot, in its present form, account for cases where the coefficient of friction is dependent on the normal load. TABOR considers the present

author's suggestions[3] and states that "... it is interesting to note that with the metals ... studied by COURTNEY-PRATT AND EISNER, where appreciable workhardening could occur, ... there was practically no variation in μ with load." On these grounds he doubts "whether workhardening can be incorporated quantitatively, in the way suggested by RUBENSTEIN, into the condition of plastic yielding of the junctions." This objection is based apparently, on a misinterpretation of the mechanism proposed in ref.[3] and the purpose of the present communication is to augment some aspects of the previous paper in an effort to remove misconceptions.

In the previous paper it was stated that "with an annealed or partially workhardened material, provided the applied shear stress remains less than the shear strength sufficiently long for the contact region material to become fully workhardened, then Amontons's first law will be obeyed. Only when the applied shear stress becomes equal to the shear strength before the contact region material has become fully workhardened will the coefficient of friction become load-dependent." Thus, whether or not the coefficient of friction depends on the load is determined not simply by whether appreciable workhardening can or cannot occur, as suggested by TABOR, but by the rate of workhardening in conjunction with the magnitude of the shear strength of the contacting regions of the metals. Unless the material is fully workhardened beforehand, it will exhibit workhardening during the application of the normal and tangential stresses and in this respect, indium and the metals studied by COURTNEY-PRATT AND EISNER[5] (gold, platinum, steel and tin) are similar. The distinctions between these metals and indium, however, are:

(i) that indium has a lower shear strength and

(ii) that indium has a lower self-annealing temperature.

As a consequence of (i) it follows that, since sliding occurs when the applied shear stress becomes equal to the shear strength of the contact region material, sliding will occur with indium at a lower applied tangential stress than in the case of the other metals. Since the work of deformation is proportional to the square of the stress, it follows that when sliding starts, less work will have been put into indium than into the harder metals.

When a metal is deformed at a temperature lower than and well removed from its self-annealing temperature, then, by definition, it is more capable of workhardening than it is when the working temperature is raised. Thus, a metal at a temperature well below its self-annealing temperature will require, in general, less working before it becomes fully workhardened than will a metal which is deformed at a temperature nearer to its self-annealing temperature. In the limit, of course, when the metal is at its self-annealing temperature, it cannot be workhardened no matter how severely it is deformed. In view of this, it follows from (ii) that indium will require more working than will the harder metals investigated by COURTNEY-PRATT AND EISNER before it becomes fully workhardened. It is evident that compared with the harder metals, indium, while requiring more deformation to become fully workhardened, will have been less heavily worked by the time sliding occurs. Hence, when sliding occurs, indium is likely to be incompletely workhardened while harder metals are

likely to be fully workhardened. Consequently, the coefficient of friction of indium is likely to be dependent on the normal load while the coefficients of friction of harder metals are likely to be independent of the normal load. It can be seen, therefore, that COURTNEY-PRATT AND EISNER's observation that the coefficients of friction of gold, platinum, steel and tin are practically independent of the normal load is consistent with the theory developed in ref.[3].

In a later section of the previous paper[3] it was suggested that if the shear strength of the contact region material were reduced by the application of a lubricant then the tangential force at which sliding commenced would be lower and the deformation and consequent workhardening of the contact regions would be less. Hence, while in the absence of lubricant the shear strength might be sufficiently high for the surfaces to remain in contact until complete workhardening occurs the reduction in the shear strength could result in sliding occurring when the contact regions are incompletely workhardened. In this way was explained the load-dependence of the coefficient of friction of copper, lubricated with fatty acids, which was reported by WHITEHEAD[6]. It will now be shown that by applying TABOR's analysis to this case the same explanation of WHITEHEAD's results follows.

TABOR[4] has shown that in the presence of good boundary lubrication $\mu \approx Si/p_0$, where Si is the shear strength of the lubricant film and p_0 is the mean normal pressure over the contact region. Now, in this case of efficient boundary lubrication, TABOR[4] has shown that junction growth in the contact region is only about 4%, *i.e.* there is relatively little interaction of the normal and tangential stresses and consequently only slight workhardening can be expected to occur before sliding begins. Thus, provided that the metal is not completely workhardened initially and provided also that the normal load is relatively low, so that only slight workhardening occurs as a result of the application of the normal load then, when sliding occurs, the metal will be incompletely workhardened. Now within the range of workhardening, the mean normal pressure, p_0, is a function of the normal load and only becomes independent of the normal load when workhardening is completed[7]. Since Si is constant it follows that μ must depend on the normal load. Thus TABOR's analysis leads to the same conclusion as that published by the present author, namely, that unlubricated metals which obey Amontons's law may, when lubricated, deviate from Amontons's law, particularly at low normal loads.

It is interesting to note that, in addition to soft metals, there is another class of materials—polymers—the coefficients of friction of which exhibit a pronounced dependence on the normal load. This dependence of the coefficient of friction on the normal load has been explained[8,9] in terms of the way in which polymers deform, namely that the deformation of these materials is in accordance with a stress(σ)–strain (ε) curve of the form $\sigma = b\varepsilon^x$ where b and x are material constants. Now this is of the same form as the stress–strain curves of soft, ductile metals in their workhardening range[10] so that the explanation of the load-dependence of the coefficient of friction of polymers is essentially the same as that put forward for soft metals; the main difference between these two classes of materials is that with soft metals the

interaction of the normal and tangential stresses is extremely important though this does not appear to be so in the case of polymers[11]. At sufficiently high normal loads the character of the deformation changes[12] and the yield stress ceases to be strain-dependent, *i.e.* x tends to zero and the coefficient of friction becomes independent of the normal load[11] (*cf.* the completely workhardened metal).

Not every case in which the coefficient of friction varies with the normal load is attributable to the influence of workhardening. For example, WHITEHEAD[6] and WILSON[13] have shown that when the coefficient of friction of a slider on a metal oxide differs appreciably from that of the slider on the metal, then increasing the normal load sufficiently to produce disruption of the oxide layer is accompanied by a change in the friction. It has been shown above, however, that in the case of soft metals, polymers and lubricated hard metals, the degree of workhardening of the material when sliding commences determines whether or not Amontons's laws of friction are obeyed. In particular, an attempt has been made to show that far from invalidating the theory published previously[3], the data quoted by TABOR[4] are consistent with it and further, that in the case of lubricated, hard metals, TABOR's analysis leads to the same conclusions as those published by the author.

ACKNOWLEDGEMENT

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Wear, 3 (1960) 150–153

Systematic Abstracts of Current Literature

Selected from literature and from Battelle Technical Review 1959

1. Deformation and Fracture

Steady-State Behavior of the Dynamic Absorber.
J. C. Snowdon. *Acoustical Society of America, Journal*, v. 31, Aug. 1959, p. 1096–1103.
The resonant vibration of machinery and resiliently mounted equipment can be reduced with a dynamic absorber of suitable design. The necessary analytical and graphical information for an optimum design is presented. It is assumed that the absorber mass is attached resiliently to the vibrating item with a rubberlike material, and not with a spring and dashpot in parallel as considered in the classical theory of Ormondroyd and Den Hartog. The dynamic absorber utilizing a material with a stiffness proportional to frequency and a constant damping factor can reduce the resonant vibration of machinery and equipment items considerably. Its performance is superior to that of the classical dynamic absorber.

The Rate of Fatigue-Crack Propagation for Two Aluminum Alloys under Completely Reversed Loading.
Walter Illg and Arthur J. McEvily, Jr. *NASA Technical Note D-52*, October 1959. 19pp., diagrs., photos, tables. OTS price, $ 0.50.
A series of crack propagation tests of sheet specimens has been conducted under completely reversed loading at various stress levels up to 30 k.s.i. Differences between effects of the compression and tension parts of the cycles are discussed. In both types of loading, the governing parameter was found to be related to the local stress at the crack tip.

Investigation of Factors Governing Fatigue Life with the Rolling-Contact Fatigue Spin Rig.
T. L. Carter, R. H. Butler, H. R. Bear and W. J. Anderson. *ASLE Transactions*, v. 1, no. 1, 1958, p. 23. For abstracts see *Wear*, v. 1, 1957/58, p. 358.

2. Adhesion and Friction

Measurements of Heat-Transfer and Friction Coefficients for Helium flowing in a Tube at Surface Temperatures up to 5900°R.
Maynard F. Taylor and Thomas A. Kirchgessner. *NASA Technical Note D-133*, October 1959. 29 pp. diagrs., photo. OTS price, $ 0.75.
Measurements of average heat-transfer and friction coefficients and local heat-transfer coefficients were made with helium flowing through electrically heated smooth tubes with length–diameter ratios of 60 and 92 for the following range of conditions: average surface temperature from 1457° to 4533°R, Reynolds number from 3230 to 60,000, heat flux up to 345,000 B.Th.U./h/ft.^{2} of heat-transfer area, and exit Mach number to 1.0. In general, the data of this investigation correlated with air data at lower temperatures obtained in previous investigations.

Effect of Lubricants on Process of Friction.
V. D. Kuznetsov, A. I. Loskutov, and L.M. Kisurina. *Henry Brutcher Translation No. 4549*, 6 pp. (From *Doklady Akademii Nauk S.S.S.R.*, v. 109, no. 1, 1956, p. 124–126.) Henry Brutcher, Altadena, Calif.
Cases in which surface-active lubricants increase (rather than decrease) the wear of parts in contact, and explanation of this behavior by the chemical interaction between lubricant and solid predominating over other variables. Results of friction tests measured by time to failure of (wire) specimens and weight loss of pulleys in contact with them.

Effect of Deformation of the Surface Texture on Rolling Resistance.
J. Halling. *British Journal of Applied Physics*, v. 10, Apr. 1959, p. 172-176.
Describes the results of a series of experiments showing the variation of the coefficient of rolling resistance, λ, with increasing load for hard steel rollers on a variety of steel surfaces. It was found that the initial value of λ, and the rate of change of λ with increasing load, were both reduced by a series of loading

cycles in which the maximum load in any given cycle exceeded the maximum load of the preceding cycle. These changes were also found to be greater for the rougher surfaces. The experimental results are interpreted in terms of the resistance to rolling being dependent on two factors; (a) the hysteresis loss concept, and (b) the requirement for the smooth roller to surmount the irregularities on the surface.

Theory of Friction of Polymers.
J. D. Huffington. The British Jute Trade Research Association. *Research*, v. 12, 1959, p. 443–446; 5 fig, 6 ref.
The multiple junction theory of friction is extended to cover a wider range of experimental conditions than previously. A new deformation index η is suggested additional to the normal index. η appears to determine the real area of contact between surfaces and is a function of the change in the distribution of asperity contacts with pressure over the region of contact. The value of η depends on whether the number of asperity contacts per unit indentation area increase (η positive) remain constant (η zero) or decrease (η negative) with increasing pressure. Maximum and minimum η values occur owing to the existence of two opposite limits in the geometrical conditions between the surfaces within the indentation area.

3. Lubrication and Lubricants

3.1. *Lubrication*

A Gas Film Lubrication Study. I. Some Theoretical Analyses of Slider Bearings. II. Numerical Solution of the Reynolds Equation for Finite Slider Bearings. III. Experimental Investigation of Pivoted Slider Bearings.
I. W. A. Gross. II. W. A. Michael. III. R. K. Brunner, J. M. Harker, K. E. Haughton, and A. G. Osterlund. *IBM Journal of Research and Development*, v. 3, no. 3, July 1959, p. 237–274.
Important characteristics of such films are determined directly from the Reynolds equation. Pressure, load, velocity, and geometry characteristics are presented for many compressible slider bearing films based upon computer solutions of a Reynolds difference equation. Cites experimental verification of computer solutions and describes experimental techniques.

A Theory of Oil Whip.
Yukio Hori. *ASME, Transactions, Series E, Journal of Applied Mechanics*, v. 26, no. 2, June 1959, p. 189–198.
Oil whip was investigated theoretically and experimentally. It is shown that the inherent instability of the rotor in previous theories can be avoided by assuming zero pressure in place of negative pressure in the oil film and that the "inertia effect" in occurrences of oil whip can be understood by distinguishing small vibrations and large vibrations of the rotor.

On the Effect of Lubricant Inertia in the Theory of Hydrodynamic Lubrication.
A. A. Milne. *ASME, Transactions, Series D, Journal of Basic Engineering*, v. 81, no. 2, June 1959, p. 239–244.
A theoretical study is made of the corrections required to the basic lubrication equations to allow for small effects of lubricant inertia in the laminar regime .The problem is approached by first determining the stream function for a wedge-shaped oil film with purely viscous flow and then obtaining the first-order correction for inertia. The effect of lubricant inertia is to cause a slight increase in the load capacity of a bearing.

Theoretical Criteria for the Effectiveness of a Lubricant Film.
E. Rabinowicz. *ASLE Transactions*, v. 1, no. 1, 1958, p. 96; 7 fig., 18 ref.
For lubricated surfaces, it is shown that only lubricants made up of very small molecules can be energetically stable on hard metal surfaces, and hence successful lubricants must form solid surface films. At the melting point of the lubricant film a transition takes place and the lubricant loses much of its effectiveness, while a second transition occurs at a higher temperature and leads to galling.

Some Considerations Relating to Lubrication of Aircraft Turbine Engines.
Alfred B. Two. *Lubrication Engineering*, v. 15, July 1959, p. 280–285.
Describes requirement of present-day aircraft engines and discusses lubricants which are currently being used.

The Lubrication of Turbine Gearing.
G. H. Clark. *Scientific Lubrication*, v. 11, June 1959, p. 12 + 8 pages.
Discusses "K" factor, scuffing, types of gear failure, pitting, tooth fracture, importance of good surface finish, and gear steels.

3.2. *Liquid lubricants*

Lubricating Oils and Greases in the Soviet Union.
N. Cheremeteff. Engineer Research and Development Laboratories, U.S. Army. May 1958. 114 pp. (Order PB 151294 from OTS, U.S. Department of Commerce, Washington 25, D.C., $2.50.)
Recent Soviet developments in the field of lubricants are reviewed in this comprehensive

study of the most up-to-date information available. The report is divided into three chapters: Lubricants; Silicons and Oil Additives; and Greases. Tables which give the characteristics of the different types of lubricating stuffs are used to supplement the text. Emphasis was placed on oil additives used to counteract wear, oxidation, mechanical deposits, unstability, poor viscosity, and results of temperature variations. Descriptions of the standard Soviet thermal stability test, corrosion test, and detergency test are presented. A table showing the chief characteristics of the internal combustion engines referred to in the text is included.

Boundary Lubrication, Wear-in and Hydrodynamic Behavior of Bearings for Liquid Metals and Other Fluids.
L. F. Coffin, Jr. *ASLE Transactions*, v. 1, no. 1, 1958, p. 139. For abstracts see *Wear*, v. 1, 1957/58, p. 357.

Thermal Conductivity of Lubricating Oils and Hydraulic Fluids.
D. W. McCready, University of Michigan Research Institute for Wright Air Development Center, U.S. Air Force. Mar. 1959. 59 pp. (Order PB 151780 from OTS, U.S. Department of Commerce, Washington 25, D.C., $ 1.50.)
An all-metal concentric cylinder type of thermal conductivity cell was designed, fabricated, and calibrated to measure the thermal conductivity of forty natural and synthetic base lubricating fluids. Thermal conductivity values in the temperature range 70-500°F are reported for fluids considered stable to the higher temperature. Such data are required for engineering designs of heat transfer equipment and were found to be unavailable from literature or other sources. The report includes complete descriptions of the design and operation of the thermal conductivity cell, stability tests used, and thermocouples.

Comparison of Temperature Effects on the Flow Properties of Greases in Capillary and in Cone and Plate Viscometers.
W. H. Bauer, A. P. Finkelstein, D. O. Shuster, and S. E. Wiberly. *NLGI Spokesman*, v. 23, no. 1, Apr. 1959, p. 15–24.
Flow properties of greases obtained with a capillary viscometer were compared with those measured by means of a cone and plane viscometer. Energies of activation for viscous flow calculated from the apparent viscosities obtained from capillary flow data differ from those calculated from the apparent viscosities given by the cone and plate viscometer. Results indicate that various structural elements in the grease have different temperature coefficients of resistance to flow, and that the extent of destruction of such elements is markedly different in the two types of viscometers.

The Structure and Lubricity of Adsorbed Fatty Films.
Yasukatsu Tamai. *Journal of Physical Chemistry*, v. 63, Aug. 1959, p. 1283–1286.
A frictional study has been made on myristic acid and cetyl alcohol adsorbed on steel surfaces by retraction from their solutions and melts. The coefficient of friction was measured in reciprocating sliding at various temperatures up to 200°C. It was found that the retraction film from solution was of some mixed-film nature, even when the solvent was non-polar. In the reciprocating friction the lubricity of the film depends largely on its fluidity which relates closely to the ability of repairing the broken part of the film during sliding, and the film gave high friction in solid state and low friction on melting. The premelting of cetyl alcohol also decreased friction to some extent.

Rate-Controlling Mechanism of Lubricating Oil Oxidation.
Foo Heng Tse. *Ohio State University, Engineering Experiment Station Bulletin* 174, Mar. 1959, 41 pp.
Development of the "film" concept and its use for explaining oxidation of lubricating oil.

Extreme Pressure Lubricants.
N. Pilpel. *Research*, v. 12, Apr. 1959, p. 141–147.
Reviews the properties of some of the materials which have been added to lubricating oils and the methods used to evaluate their effectiveness.

Chemical Additives Control of Frictional Characteristics of Lubricants.
S. R. Sprague and R. G. Cunningham. *I.E.C.* v. 51, Part 1, Sept. 1959, p. 1047; 7 fig., 3 ref.
Special antifriction lubricants are required for noise-free operation of mechanisms containing wet clutches, such as automatic transmissions. Performance characteristics that are affected by the frictional nature of the lubricant include clutch lockup duration and shock, torque capacity, wear and fading tendencies, and audible frictional vibrations. A laboratory friction test apparatus (modified four-ball wear tester) has been utilized in studies of the several parameters that influence the design of practical power transmission fluids. Experimental data illustrate effects of antifriction additive type and concentration, interaction of additives, lubricant degradation, and clutch facing materials.

Designing Oils for a 2-Stroke Engine.
J. W. Savin. *SAE Journal*, v. 67, Aug. 1959, p. 62–63.
Fundamental requirements of a lubricating oil; lab test of oils.

Effects of Lube Oils on 2-Stroke Engines.
H. V. Lowther, G. H. Shea, and K. C. Kresge. *SAE Journal*, v. 67, July 1959, p. 72–74.
Laboratory studies show effect of detergents on piston cleanliness, effect of deposition of solid combustion products in the exhaust ports, engine wear and spark-plugs.

Lubrication of Large Piston Compressors. (in German)
G. Westhoff. *Chemie-Ingenieur-Technik*, v. 31, 1959, p. 516.
As is the case with many other machines, it is necessary in the case of piston compressors to distinguish between the drive gear lubrication and the piston lubrication. Only in the case of low-pressures is it possible to use the same oil for both purposes. Under the diverse operating conditions in which piston compressors must operate in the chemical industry demands are made on the driving gear lubricant and the cylinder lubricant which are so different that two types of oil are necessary. Practical experience in the selection of lubricants is reported.

Further Pump Tests on GE. 81406 Synthetic Hydraulic Fluid.
B. Perks and J. R. Wheeler. Royal Aircraft Establishment (Gt. Brit.) May 1959. *RAE Tech. Note Mech. Eng. 291*, 11 pp., diagrs., tables (Ask for CN-76109*).
G. E. 81406 synthetic hydraulic fluid was subject to a 1,000-hour endurance pump test at 4,000 p.s.i. delivery pressure and 100°C fluid temperature after a "run-in" period at reduced load. The conditions and results of the test are outlined in this note and wear figures are compared with those obtained from a 500-hour run with D.T.D. 585 fluid.

3.3. *Solid lubricants*

Solid Film Lubrication at High Temperatures.
E. P. Kingsbury. *ASLE Transactions*, v. 1, no. 1 1958, p. 121. For abstracts see *Wear*, v. 1, 1957/58, p. 357.

Effect of Solid Additives on Fluid Lubricants.
Harold L. Herzig. *Lubrication Engineering*, v. 15, Apr. 1959, p. 152–158.
The effect of solid additives in fluid lubricants on the coefficient of friction is attributed to the ability of additives to smooth the surface of a bearing during operation and to the shear strength of the additives. By proposed empirical formulation, the effects of particle dimension and concentration of the additive, and the effect of error in geometry of the bearing are quantitatively described. Data from a foil-bearing experiment support the formulation and confirm the potential of molybdenum disulfide as an additive to fluid lubricants.

Investigation of the Mechanism of Tungsten Disulfide Lubrication in Vacuum.
M. T. Lavik, G. E. Gross, and G. W. Vaughn. *Lubrication Engineering*, v. 15, June 1959, p. 246 + 4 pages.
Lubrication behavior was found to be similar to that of MoS_2. The steady-state value for the coefficient of friction at room temperature is 0.065. When sliding is resumed, following a "stop-time", the coefficient of friction is initially high. It drops rapidly, however, and approaches the equilibrium value within a few minutes.

4. Bearings, Gears and Related Machine Parts

Bearing and Seal Development.
Stanley Gray. *Mechanical Engineering*, v. 81, Apr. 1959, p. 76–80.
Some current research by an aircraft-accessory manufacturer on bearings for operation up to 140,000 r.p.m. and 100°F, and on the unusual lubricants and seals needed.

Recent Developments in Roll Neck Bearing Design and Mounting.
E. C. Denne, Jr. *Iron & Steel Engineer*, v. 36, no. 5, May 1959, p. 117–126.
The multi-row cylindrical roll neck mounting consists of two separate parts, a 4-row cylindrical radial bearing for the heavy mill separating loads and a separate thrust bearing for the axial load. Describes some of the many possible multi-row and thrust bearing combinations which satisfy the most exacting mill requirements.

On Critical Speeds of a Shaft Supported by a Ball Bearing.
Toshio Yamamoto. *ASME, Transactions, Series E, Journal of Applied Mechanics*, v. 26, no. 2, June 1959, p. 199–204.
Speeds have peculiar modes of vibrations which are determined by dimensions of ball bearings; one of them is motion of forward precession, the other is backward precession. Describes the cause of these critical speeds and the behavior of those vibrations.

Energy Losses of Balls Rolling on Plates.
R. C. Drutowski. *ASME, Transactions, Series D, Journal of Basic Engineering*, v. 81, no. 2, June 1959, p. 233–238.
An apparatus for measuring the rolling force of a ball supported between two plates is described. Instantaneous values of the rolling force vary greatly from point to point on the sample surface and this variation is explained

in terms of surface roughness and material homogeneity. The energy losses are shown as functions of load, material, and surface roughness.

Ball Bearing With Near-Zero Torque Upgrades Gyro Performance.
Military Systems Design, v. 3, no. 2, Mar.-Apr. 1959, p. 68.
The "Dynamic" bearing is a self-contained, electromagnetically powered ball-bearing assembly that closely approaches the minimum friction achieved by knife edges, jeweled pivots or flotation devices, yet provides rugged support for relatively heavy gyro suspensions.

Elastic and Damping Properties of Cylindrical Journal Bearings.
B. Sternlicht. *ASME, Transactions, Series D, Journal of Basic Engineering*, v. 81, no. 2, June 1959, p. 101–108.
Derivation of equations; solution with arbitrary direction for the squeeze-film velocity; techniques for calculation of fluid stiffness and dissipation functions.

A Solution for the Finite Journal Bearing and its Application to Analysis and Design-I-II-III.
A. A. Raimondi and J. A. Boyd. *ASLE Transactions*, v. 1, no. 1, 1958, p. 159–194. For abstracts see *Wear*, v. 1, 1957/58, p. 358.

The Determination of the Characteristics of Hydrostatic Bearings through the Use of the Electric Analog Field Plotter.
A. M. Loeb. *ASLE Transactions*, v. 1, no. 1, 1958, p. 217. For Abstracts see *Wear*, v. 1, 1957/58, p. 358.

Railroad Car Journal Plain Bearings.
Lubrication, v. 45, Aug. 1959, p. 101–116.
Design, construction, characteristics, testing, wicking, and lubricants.

Review of Propellor Shaft Thrust Bearings.
B. Sternlicht, J. C. Reid, Jr., and E. B. Arwas. *American Society of Naval Engineers, Journal*, v. 71, no. 2, May 1959, p. 277–289.
Reviews operating conditions of propeller shaft thrust bearings and present status of bearing technology and suggests areas of investigation necessary if thrust bearings are to keep up with advancing requirements of marine propulsion machinery.

Characteristics of a Self-Lubricated Stepped Thrust Pad of Infinite Width With Compressible Lubricant.
Kikuo C. Kochi. *ASME, Transactions, Series D, Journal of Basic Engineering*, v. 81, no. 2, June 1959, p. 135–146.
Harrison's equation for the pressure in a gas-lubricated bearing of infinite width is solved for a thrust pad with stepped configuration. Analytic expressions for the pressure and load are developed. Numerical results are presented graphically. Determination of optimum design parameters is given by a pair of graphs.

Surface Finish and Clearance Effects on Journal-Bearing Load Capacity and Friction.
F. W. Ocvirk and G. B. Dubois. *ASME, Transactions, Series D, Journal of Basic Engineering*, v. 81, no. 2, June 1959, p. 245–253.
A method of relating surface finish to minimum oil-film thickness and the corresponding load capacity of plain journal bearings is presented with supporting experimental data. The effect of clearance on load capacity and friction are shown on graphs indicating an optimum bearing clearance.

Axial, Relative Motion of a Circular Step Bearing.
L. Licht. *ASME, Transactions, Series D, Journal of Basic Engineering*, v. 81, no. 2, June 1959, p. 109–117.
Equations relating the flow of the lubricant and the axial motion of an externally pressurized thrust bearing are developed. Bearing is stable when the fluid is incompressible. Expressions for local stiffness and damping coefficients given, an analog computer solution of the equation of motion is compared with the results of the corresponding, small displacement equation.

5. Wear and Wear Resistance

5.1. *Materials*

Investigation of Ignition Temperatures of Solid Metals.
W. C. Reynolds, Stanford University. *NASA Technical Note D-182*, October 1959. 71 pp., diagrs., photos., tables. OTS price, $ 2.00.
The ignition temperature of a solid metal is related through a thermal definition of ignition to the rate of oxidation and to the radiation and convection heat-transfer parameters. The mechanisms of oxidation are reviewed and the factors which influence ignition temperatures are discussed. Reasonable agreement between theoretical and experimental ignition temperatures is demonstrated. Experimental ignition temperatures for several metals are presented.

The Role of Atmospheric Oxidation in High Speed Sliding Phenomena II.
M. Cocks. *ASLE Transactions*, v. 1, no. 1, 1958, p. 101. For abstracts see *Wear*, v. 1, 1957/58, p. 360.

A Fundamental Study of Synthetic Sapphire as a Bearing Material.
L. F. Coffin, Jr. *ASLE Transactions*, v. 1,

no. 1, 1958, p. 108; Discussion: see *ASLE Transactions*, v. 1, no. 2, 1958, p. 345–346.
Experiments are reported on the sliding behavior of synthetic sapphire against certain fairly pure metals as a function of temperature. The initial selection of the sapphire–metal couple was determined on the basis of thermodynamic and physical consideration so as to produce, theoretically, a weak interfacial contact region. It was observed that this basis for selection of desirable sapphire–metal couples needed to be modified to include the effect of the surrounding atmosphere.
Experiments on the sliding characteristics of these couples as a function of both high and low temperatures are included and discussed.

Friction and Wear of Metals to 1000°C.
E. P. Kingsbury and E. Rabinowicz. *ASME, Transactions, Series D, Journal of Basic Engineering*, v. 81, no. 2, June 1959, p. 118–122.
Sliding experiments using the metals Zn, Cu, Ti, and 1020 steel were carried out on a pin-on-disk friction apparatus incorporated in a metallurgical furnace.

Chrome Face vs. Iron Side Wear – An Analysis of Some Radioactive Piston-ring Wear Studies.
R. G. Abowd. *ASLE Transactions*, v. 1, no. 1, 1958, p. 91. For abstracts see *Wear*, v. 1, 1957/58, p. 360.

Austenitizing Behavior and Rapid Life Test of Drawn Wires for Bearing Balls.
Manabu Ueno. *National Research Institute for Metals, Transactions*, v. 1, no. 1, 1959, p. 22–26.
The austenitizing behavior and rapid life test property of five kinds of wire, used for balls, rollers, and needles of bearing, were investigated. The effect of ingot size on the lifetime was not appreciated, but the effect of different raw materials on it was apparent. The lifetime became better but crack damage worse on account of the V addition.

Martensitic White Irons for Abrasion-Resistant Castings.
T. E. Norman, A. Solomon, and D. V. Doane. *Modern Castings*, v. 35, Apr. 1959, p. 104–118.
This paper deals with a number of the important characteristics of the martensitic white irons, together with studies on a number of the more important variables influencing their structure, wear resistance, mechanical properties, and resultant service life.

Anti-Friction Coatings of Sprayed Pseudo-Alloys.
L. V. Krasnichenko. *Metal Industry*, v. 94, June 5, 1959, p. 461–463.
Development of new anti-friction materials to replace bearing bronzes and other bearing alloys in sliding bearings by making use of the electric metal spraying process.

New Retainer for Aircraft Gas Turbine Bearings.
F. C. Wagner and J. T. Burwell, Jr. *ASLE Transactions*, v. 1, no. 1, 1958, p. 1; 3 fig., 3 tables, 15 ref.
A study of the wear behavior of several classes of potential cage materials for aircraft gas turbine bearings was conducted, using a special wear-testing machine to approximate the projected service conditions of such bearings. Several heterogeneous alloys containing silver as a soft, low shear strength phase showed superior wear properties to presently used silicon-containing nickel–copper alloy ("S"-monel) and iron silicon bronze, even when silver-plated. The latter alloys were used as reference cage materials.
The hard, load-supporting phase showing the most promise was a nickel base with additives of silicon or a silicide. The materials showing the best wear properties were most efficiently fabricated by powder metallurgy techniques, utilizing the infiltration method for adding the silver.

Analysis of Bearing Material for Elevated-Temperature Application.
J. P. Denny. *U. S. Atomic Energy Commission APEX-464*, Aug. 1957, 38 pp.
Promising areas of investigation are specified for the development of bearing materials which will operate in air at elevated temperatures. A discussion of physical and chemical properties of bearing materials and of basic principles to be considered in this development is followed by recommendations.

Wear of Carbon-type Seal Materials with Varied Graphite Content.
Max A. Swikert and Robert L. Johnson. *ASLE Transactions*, v. 1, no. 1, 1958, p. 115; 8 fig., 4 ref.
Materials having graphite carbon as the major constituent were studied as possible seal materials at 10,000 feet per minute sliding velocities; in most experiments the temperature of the mating surfaces was 500°F. Carbon materials made graphitic by electrographitization were too soft; they gave high wear and high friction. Bodies molded with high-graphite-content materials and made hard by improved molding methods and impregnation gave acceptable friction and wear properties.

Wear Trials on Resin-Rubber Soles.
D. L. Clarkson. *Rubber and Plastics Age*, v. 39, no. 9, Sept. 1958, p. 751.
Soles of natural and synthetic rubbers with high styrene resin, PVC soles, and leather soles were tested for durability and spread. Results of the tests led to a formulation using Butaker S 7001 and natural rubber, which showed the highest durability and no appreciable spread in tests over a 12-month period.

5.2. Wear resistance

Investigation of High-speed Impact: Regions of Impact and Impact at Oblique Angles.
James L. Summers. *NASA Technical Note D-94*, October 1959. 18 pp., diagrs., photos. OTS price, $ 0.50.
Small metal spheres of various densities were fired into copper and lead targets at impact velocities to 11,000 feet per second. The regions or types of impact are described. In the fluid-impact region, the target penetrations and cavity volumes are correlated on the basis of the ratio of impact velocity to speed of sound in target material and the ratio of projectile to target density. A correlation of penetration for normal and oblique impact is presented.

A Photographic Study of the Impact Between Water Drops and a Surface Moving at High Speed.
D. C. Jenkins and J. D. Booker. Royal Aircraft Establishment (Gt. Brit.). *RAE Tech. Note Mech. Eng. 275* (Ask for CN-75611*).
The normal impact between 2-mm-diam. water drops and a smooth, hard, surface moving at 1,000 ft/sec has been studied photographically and the results discussed. The speed of the radial flow resulting from the impact has been measured and an estimate made of the corresponding pressure existing between the drop and the surface. The case of impact of drops on rough, deformable, and inclined surfaces has also been briefly considered.

The Effect of Lubricants on Gear-tooth Surface Fatigue.
T. F. Davidson and P. M. Ku. *ASLE Transactions*, v. 1, no. 1, 1958, p. 40. For abstracts see *Wear*, v. 1, 1957/58, p. 358.

The Effect of Aircraft Gas Turbine Oils on Roller Bearing Fatigue Life.
M. E. Otterbein. *ASLE Transactions*, v. 1, no. 1, 1958, p. 33. For abstracts see *Wear*, v. 1, 1957/58, p. 357.

6. Analysis and Testing

Gas Chromatographic Determination of Dissolved Oxygen in Lubricating Oil.
Paul G. Elsey. *Analytical Chemistry*, v. 31, May 1959, p. 869–870.
Method developed in connection with a program relating engine wear to the amount of oxygen present in lubricating oil.

Radioactive Determination of Bearing Wear in a Refrigeration Compressor.
G. R. Fox and R. C. Elwell. *Lubrication Engineering*, v. 15, Apr. 1959, p. 144–151.
Theoretical load-carrying capacity at different operating conditions is compared to actual bearing wear determined by radioactive methods. This technique allows a continuous and rapid evaluation of a complete fluid film bearing system.

Améliorations dans l'utilisation des radiotraceurs pour l'étude de l'usure des segments de piston méthode IFP-RA3-bilan d'usure.
Improvements in the Use of Radiotracers for Studying Wear in Piston Segments.
Jean Thiéry. *International Journal of Applied Radiation and Isotopes*, v. 5, no. 3, May 1959, p. 197–203 + 1 plate.
By collecting the oil moving down to the sump in a gutter at the bottom of the liner, a great improvement in sensitivity is obtained. Difference between the activity of this oil and that of the sump oil, by which the ring belt area is fed, is continuously registered. The curve obtained represents variations of instantaneous wear-rate, and by integrating the graph of oil volume versus activity, one can obtain the "apparent wear", as carried in the oil.

Low-Cost Engines for Lubricants Testing.
Norman Kendall. *SAE Journal*, v. 67, July 1959, p. 64–65.
Describes three inexpensive engines for testing and evaluating lubricating oils.

Cermet Body Nondestructive Test Data Evaluated by Spin Tests of Blades and Bend Tests of Bars.
A. G. Holms and A. J. Repko. *Non-destructive Testing*, v. 17, no. 3, May-June 1959, p. 156–164.
High-quality cermet blades, previously screened by surface inspections for flaws, were re-inspected at the root by means of several flaw-detection and property-measurement tests. Strengths of these roots were determined by short-time room-temperature spin testing to fracture.

The Wright Air Development Center Gear Lubrication Research and Development Program.
H. R. Smith. *Lubrication Engineering*, v. 15, July 1959, p. 290–294.
Outline of programs presently underway. Apparatus and test procedures for determining scuff limited load and gear fatigue; effect of operating variables on test procedures. Plans for future work.

7. Surface Treatment and Finishing

The Nature of Industrially Polished Metal Surfaces.
L. E. Samuels. *Australian Institute of Metals, Journal*, v. 4, no. 1, May 1959, p. 1–11.
See also *Electroplating and Metal Finishing*, v. 12, 1959, p. 130–133.
Industrial polishing occurs basically by cutting and not by surface flow. Deformed layers are produced on the surface. Quantitative data on the depth of the layer are given for brass specimens finished by a representative range of industrial methods. A number of associated surface effects are considered together with their likely effects on electrodeposits subsequently applied to the surface. Of particular interest is the observation that the obliteration of surface cavities, which is frequently noted after industrial polishing, appears to be due to the collapse of the cavities by compression and not to the smearing of a film across the surface.

Hardsurfacing Can Increase Life of Tool Joints and Drill Collars.
Raymond R. Fayer. *World Oil*, v. 149, Aug. 1959, p. 68–71.
Describes recommended pattern for surfacing.

8. Machining and Tool Wear

A Theory of Cutting-tool Wear and Cutting-oil Action.
A. A. Dorinson. *ASLE Transactions*, v. 1, no. 1, 1958, p. 131. For abstracts see *Wear*, v. 1, 1957/58, p. 361.

A Three-Dimensional, Tool-Life Equation—Machining Economics.
Bertil N. Colding. *ASME Transactions, Series B, Journal of Engineering for Industry*, v. 81, no. 3, Aug. 1959, p. 239–250.
Two tool-life equations are derived, one limited equation and one general tool-life equation, between the variables cutting speed, chip equivalent, and tool life. The chip equivalent, introduced by Woxén, is a well-defined function of feed, depth of cut, nose radius, and side-cutting-edge angle. The limited equation takes into account the variation of Taylor's exponent n with the value of the chip equivalent, but the equation is only valid within certain limits of cutting speed and chip equivalent. An expression called the productivity is derived. This relationship is valid for either maximum production or minimum cost and, combined with the general, hyperbolic, tool-life equation, it is used to investigate the optimum combination of tool-life, cutting speed, and chip equivalent.

Chip Formation During the Turning Operation in the Presence of a Built-Up Nose.
Bernard W. Shaffer. Paper from *Proceedings of the Third U.S. National Congress of Applied Mechanics*, The American Society of Mechanical Engineers, New York, 1958, p. 655–664.
Chip formation in the turning operation, when some of the workpiece adheres to the base of the cutting tool forming a built-up nose, is analyzed as a plane strain problem of a perfectly plastic material. A slipline field is presented which satisfies the stress and velocity boundary conditions of the problem. The results of this solution are then applied to provide expressions for the machining force and chip-thickness ratio. The solution for the turning operation is found to approach that for the orthogonal planing operation, as the radius of the workpiece is increased.

Factors Influencing the Performance of Grinding Wheels.
E. J. Krabacher. *ASME, Transactions, Series B, Journal of Engineering for Industry*, v. 81, no. 3, Aug. 1959, p. 187-200.
A study of the nature of grinding-wheel wear indicates that the grinding-wheel wear curve is similar to those of other cutting tools. It demonstrates further that the type of grinding operation significantly affects the nature of wheel wear. A unique technique was developed for very accurately measuring grinding-wheel wear. This measured wear may be translated into terms of "grinding ratio", which is the generally accepted parameter for measuring wheel wear. It is the ratio of the volume of metal removed per unit volume of wheel worn away.

The Wear of Abrasives in Grinding.
G. J. Goepfert and Josephine L. Williams. *Mechanical Engineering*, v. 81, Apr. 1959, p. 69-73.
Chemical reactions between metal and abrasive, their effect on abrasive wear, and modification by grinding fluids.

Bibliographies

Friction and Wear in Machines, No. 13 (in Russian)*

Academy of Sciences U.S.S.R., Moscow, 1959; 266 pp.; 11 Roubles.

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Book Reviews

Friction and Wear, edited by R. DAVIES, Elsevier Publishing Company, Amsterdam, 1959. $6\frac{1}{2} \times 9\frac{1}{2}$ in., viii + 191 pp., 120 illus., 13 tables; £ 1.16.6.

This book constitutes the proceedings of the first of a series of annual symposia organized by the Research Laboratories of General Motors Co. and held near Dearborn, Michigan, in June, 1947. This particular symposium was devoted to friction and wear because, as the editor explains in his preface, recent developments in this field have significantly influenced modern technical progress. If the reviewer is right in his contention that these symposia aim at providing opportunities for assembling and exchanging ideas in fields that are influenced by a variety of scientific disciplines, the present field constitutes indeed a suitable and timely choice.

Various leading authorities on friction and wear have presented the following contributions in a generally laudable way:

R. DAVIES, "A Tentative Model for the Mechanical Wear Process"; R. C. DRUTOWSKI, "Energy Losses of Balls Rolling on Plates"; L. F. COFFIN, JR., "Some Metallurgical Aspects of Friction and Wear"; C. L. Goodzeit, "The Seizure of Metal Pairs during Boundary Lubrication"; F. P. BOWDEN "Recent Experimental Studies of Solid Friction"; W. A. ZISMAN, "Friction, Durability and Wettability Properties of Monomolecular Films on Solids"; E. RABINOWICZ, "A Study of the Stick-Slip Process"; A. T. GWATHMEY AND L. D. DYER, "Cohesion between Two Oriented Single Crystals of Copper".

H.B.

Internal Stresses and Fatigue in Metals, edited by G. M. RASSWEILER AND W. L. GRUBE, Elsevier Publishing Company, Amsterdam, 1959. $6\frac{1}{2} \times 9\frac{1}{2}$ in., x + 451 pp., 280 illus., 9 tables; £ 3.12.0.

In this book are presented the papers read at a symposium on internal stresses and fatigue in metals, which was held at General Motors Technical Center (Detroit and Warren, Michigan, U.S.A.) in September 1958. The phenomena of internal (= residual) stresses and fatigue are considered —in some cases even in interrelation—using the atomistic approach of the solid state physicist or the phenomenological point of view of the engineer.

Physicists interrelate dislocations (involving stress fields) and fatigue. Some of the papers are of a theoretical nature; the majority relate to experimental work. In theoretical contributions ESHELBY, HAASEN AND FRIEDEL consider dislocation stress fields, also in interaction. In the experimental sector, results are described of various experimental techniques, *viz.* light microscopy (HIRSCH describes direct observations of dislocations made by transmitting directly thin metal specimens of about 200 Å thickness), X-ray microscopy, field ion microscopy, birefringence in infrared polarized light, and X-ray diffraction methods. Some experiments performed at whiskers are described. In the phenomenon of corrosion of steel, whisker growth appears to be a consistent process; this was investigated by means of electron microscopy, transmitting the edge of small specimens. Electron interference effects (Moiré patterns) in thin crystals are described; one of the illustrations is a remarkable electron micrograph of a microcrack (Fig. 5 on p. 124). Nucleation and growth of fatigue cracks are described, placing emphasis on FORSYTH's discovery of extrusions and intrusions.

Engineers have correlated internal stresses and fatigue strength (DOLAN, HEMPEL, MATTSON AND ROBERTS). The influence of internal stress appears to be limited to low stress levels, since at high levels of plastic strain range any internal strains are washed out. In the paper of COFFIN, one might prefer to read "total strain range" instead of "plastic strain range" in Figs. 14–16, since at low load the elastic part of the deformation is not negligible. In other papers the surface condition is considered, shot peening being one of the topics discussed. Further subjects dealt with are: phase transformations, *e.g.* those caused by grinding (KOISTINEN), effect of environment, and hydrogen diffusion in steel.

Especially remarkable is the subject of measuring internal stress. A measuring method employing X-ray diffraction is described (GUINIER). This technique was applied to hardened steel by MARBURGER AND KOISTINEN.

In the field of fatigue a change of focus appears on internal stress; this book gives an excellent picture of the present stage of the development in this direction. It is up-to-date, appearing within a year of the symposium taking place, highly stimulating, illustrated effectively and, also typographically, of a high standard; it can be recommended to research workers who are interested in the fundamental problems of fatigue.

J.L.

Notes on Contributors

J. U. AUGUSTIN: geboren in Bonn/Rhein; Studium in Bonn und Karlsruhe; erwarb in 1920 den Dipl.-Ing. und Oberreg.-Baurat. Nachher 10 Jahre in der Industrie, 12 Jahre in dem Entwicklungsamt für Heeresmotorisierung, 10 Jahre an der T. H. Dresden (1928 u. 1929) sowie ab 1950 an der T.U.-Charlottenburg. Spez: Kraftfahrzeugbau und in Verbindung hiermit Untersuchungen auf dem Gebiete der Reibung und des Verschleisses.

[See p. 114]

IAIN FINNIE: born in Hong Kong; educated in Hong Kong, Scotland and Canada; technical education at University of Glasgow, Scotland, B.Sc. (1st hons.) and Massachusetts Institute of Technology, S.M., M.E., and Sc.D.; doctoral thesis at M.I.T on the shear and friction processes in the cutting of metals. Employed as instructor at M.I.T. 1952-1953, and subsequently as engineer in the Materials Engineering and Corrosion department of the Shell Development Company, Emeryville, California. Author of a number of technical papers on metal cutting, erosion and creep of metals. Coauthor of the book *Creep of Engineering Materials* (McGraw-Hill Book Co., 1959).

[See p. 87]

C. RUBENSTEIN: for biographical note see *Wear*, *2* (1958/59) 165.

[See p. 150]

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[See p. 122]

G. ULSENHEIMER, Dr. rer. nat.: geboren in Schweinfurt (Deutschland); Studium der Physik an der Universität Göttingen und T.H. München; Diplom in Physik 1939; Forschungslabor der Fa. Kugelfischer Georg Schäfer & Co., Schweinfurt 1939–1945; Dissertation an der T.H. München 1950; seitdem Institut für Technische Mechanik der T.H. München. Verstorben am 26. August 1959 in München.

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